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OF HEATING AND VENTILATING
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FALL MEETING
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AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

No. 1144

FORTY-SIXTH ANNUAL MEETING, 1940
Cleveland, Ohio

WITH an attendance that approached an all-time record the 46th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS was held in the Hoted Statler, Cleveland, Ohio, January 22 to 26. In a total registration of over 1,100 members, ladies and guests, it was notable that over one-fifth of the total Society membership was registered at this meeting. The members were attracted by an important group of committee meetings, a fine technical program, and the 6th International Heating and Ventilating Exposition. The turnout was such that the city's hotel facilities were taxed to the limit.

The 46th Annual Meeting was called to order by Pres. J. F. McIntire in the grand ballroom of the Hotel Statler, Cleveland, Tuesday morning, January 23. Hon. Harold L. Burton, mayor of the City of Cleveland, welcomed the members and said:

"We appreciate your coming as it enables our engineers in this part of the country to share the benefits that you bring in advancing the interests of heating, ventilating, and air conditioning."

He also said, "In these difficult times of the depression, I have often thought that although the times are hard and although we go through great difficulties of one kind or another, there are a great many things that are helpful to us, and I believe that a part of that help is in the pressure that is being put upon us to develop new mechanical and scientific appliances and devices."

He said further, "There are so many more foci of development in our generation than there ever were before; there are so many more places to develop the man-made wealth than there ever have been before, that there is a guarantee that we will develop it. There is much more knowledge available because no longer does a man invent by himself, but he has the wealth of the world; the translated knowledge of the world is his research bureau to help him about it. An inventor is no longer a man unto himself, but a man with the help of all the world at his hand to help him do the inventing and advancing. The means of obtaining materials and distributing products are so much more improved that, personally, I believe if we will

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but let ourselves use them, this lag between new invention and unemployment that follows can be cut down so that we will in our generation be able to move up the advance from one invention to another to a much closer period and thereby find in it a much greater amount of wealth available to us than before.

"I want to illustrate my point by the radio which, of course, 20 years ago was unknown. Today it is with us with 340,000 employees in that industry where there were none 20 years ago. That means a million and a half people dependent upon a new mechanical device and scientific advance.

"The same thing is true in your own field. You are moving into a field that has never been fully developed or properly understood in the nation. It is a field, therefore, in which we have an opportunity by realizing that by the combination of our efforts and our knowledge we are able to move into a field of wealth, a standard of living that has not been available to us before.

"And so as you come here we welcome this industry that is advancing into that new field as you are. We welcome you as a city that has been largely publicized in the past few months because of its relief troubles. That does not mean that the city is suffering an industrial collapse at all. It is just an indication of the psychology that sometimes wrongly follows a bit of publicity.

"We are moving ahead here. We are feeling the effect of an advance in industry, of an advance in new products and we welcome here an industry that will help us to look into the future and create needed added wealth that can be added to our standard of living.

"I hope that you enjoy your stay here with us. We enjoy having you with us. We want to do everything possible to make your stay not only pleasant but productive to you."

President McIntire responded briefly and expressed the appreciation of the Society for the cordial welcome given.

In a brief resumé of his administration he pointed out that the accomplishments of the past 12 months would not have been possible without the work that had been done in previous years, so that the progress of the Society was cumulative. He paid tribute to the work of the Guide Publication Committee in producing an outstanding volume that contains more technical data in a smaller book. He mentioned the progress of the Society's research work and the excellent financial condition of the Society. He complimented the Meetings Committee for the fine programs that had been arranged for the general meetings of the Society, and stated that the Standards Committee was making progress in establishing a logical procedure for the development and administration of codes. President McIntire also mentioned the excellent work being done by the chapters and the value of the Chapter Delegates' Conference held at each annual meeting, and in conclusion he expressed appreciation to past presidents for their cooperation and assistance during his administration.

The report of the secretary was given by A. V. Hutchinson as follows:

Report of Secretary

During 1939 the Society gained much in prestige and it has definitely extended its sphere of influence professionally as well as geographically. With a current membership of 3147, and local chapters in 29 principal cities, it requires a New York headquarters office staff of 10, and a research staff of 19 in the Laboratory at Pittsburgh, as well as the cooperative efforts of several hundred members serving on committees and in chapters to carry on the activities sponsored by the Society.

A brief review of the work at the headquarters office will give some idea of the various duties required to administer the financial, membership, publication, meetings, employment, chapters and committees.

In the 300 working days of the calendar year it is necessary to carry on certain

routine matters specified by the Constitution and By-Laws, as well as various special projects sponsored by the Council and Committees.

Some of the accomplishments of the past year may be mentioned as follows:

1. Collection and deposit of dues, research pledges, publication and other miscellaneous income and disbursements of budgeted expenditures of $97,650.00 for general Society accounts and $27,936.00 for research. Sending quarterly dues bills and monthly statements on miscellaneous accounts.

2. Keeping the Roll of Membership up-to-date; advising chapters and Journal publishers of changes of addresses. Carrying on the procedure of membership, elections totalling 388, and cancellations, as directed by the Council.

3. Securing papers, editing manuscripts, preparing notes and discussions for the monthly JOURNAL and TRANSACTIONS. Handling the sales promotion campaigns for the sale and distribution of 11,000 copies of THE GUIDE 1939, and sale of catalog data space for 1940 edition, as well as styling and coordinating and indexing 46 chapters of the Text Section.

4. For the first time three general meetings of the Society were held during one year, respectively, in Pittsburgh, Pa., Mackinac Island, Mich., and Atlanta, Ga. This plan permitted the attendance of a great many members from different sections of the country, and the result of holding three meetings was most gratifying because the total registration of members and guests was close to 1000.

5. Employment file was maintained for the benefit of members and a great many placements were recorded during the year.

In the membership service work assistance was given to the Local Chapters by the Speakers Bureau Committee and 15 assignments were completed and, in addition to making the preliminary arrangements, advance publicity material, and other essential correspondence was required in this activity. Assistance was given by the Secretary's office in the organization of three new Chapters during the year, namely, North Carolina, Oregon and Delta Chapter at New Orleans.

It should be noted in the report of the Finance Committee that the cost of administering membership activities amounted to $1.25 per member in excess of income, but a surplus from Guide operations permitted an increase in the General Fund of the Society for the year 1939, so that it is in excellent financial condition and membership is gaining in strength.

Respectfully yours,
A. V. HUTCHINSON, *Secretary.*

A. J. Offner, New York, chairman of the Finance Committee read his report which summarized the audit of the Society by the Certified Public Accountant.

Report of Finance Committee

In submitting the financial report for the calendar year 1939, your Finance Committee is pleased to advise that the finances of the Society continue in an excellent and sound condition.

The accounts of the Society have been audited by a Certified Public Accountant as required by the By-Laws. The Balance Sheet shows that the net assets of the Society, including Research, amount to $112,923.38. Of this total there is $66,081.54 in the operating and surplus accounts and $30,596.80 in the Reserve and Endowment Funds and invested in securities of the United States Government. Comparing the total income and expenditures of the Society during the past year with the estimated figures set up in the Budget as adopted by the Council at the beginning of the year, it is found that the income was about $5,000 under estimates, while expenses were nearly $14,000 less than Budget provisions. The operations of general Society activities show a loss for the year, but with the surplus from GUIDE operations and deductions of reasonable reserves leaves an addition to the general fund amounting to $1,642.87. Research activities had an excess of $7,626.38 in expenditures over current income, the difference being made up by reserve funds.

The detailed statement of the Society's assets and liabilities as prepared by the

Society's Certified Public Accountants, Tusa and La Bella, is attached to this report and will be published in the Society's JOURNAL.

Subject to the requirements of the Constitution, 40 per cent of the dues collected from Members and Associates amounting to $14,903.73 was turned over to the Committee on Research. In addition to this amount the Council appropriated $5,000, making a total of $19,903.73 contributed by the Society during the past year for Research work. The total amount spent for Research in 1939 was $44,262.39, which in addition to Society contributions includes earmarked contributions, unearmarked contributions and contracts. A detailed financial statement of the Committee on Research will be presented by the Chairman of that Committee.

As a part of its duties, and with the approval of the Council, the Finance Committee sold above par $4,000 par value U. S. Treasury Bonds due in 1941. This sale was made to protect the financial interest of the Society due to the nearness of the maturity date and the general tendency for such bonds to gradually drop to par. The present market value for this series of Treasury Bonds has proven the soundness of disposing of them. The Constitution and By-Laws of the Society limits the investment of surplus Society funds to U. S. Government Securities and those securities which are legal for savings banks in the State of New York. During the past year, the Finance Committee, with Council approval, has invested $7,500, the limit permitted by law, in U. S. Savings Bonds of $10,000 par value at maturity date. This will give an interest return at maturity of 50 per cent more than could have been obtained had this money remained in savings banks accounts. These bonds can be sold at any time at the original cost plus interest to date of sale. Any further surplus funds available will be invested at opportune time for sound and legal investments. Pending this opportunity your Committee recommends holding a cash reserve in savings banks accounts.

The constitution and By-Laws require that the general accounts and the Research accounts of the Society be segregated and be kept separate. During the past year, the books and accounts formerly kept at Pittsburgh have been transferred to New York and are now being handled by the office staff in New York.

Respectfully submitted,
A. J. OFFNER, *Chairman*
Finance Committee.

Report of Certified Public Accountant

January 18, 1940

American Society of Heating and
    Ventilating Engineers,
51 Madison Avenue,
New York, N. Y.
Gentlemen:

Pursuant to your request, we made an examination of the books of account and records of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS—New York, N. Y., and the Research Fund for the calendar year ended December 31, 1939, and submit herewith our report.

The work covered a verification of the assets and liabilities as of the date previously stated and a review of the operating accounts for the year ended December 31, 1939. For the period recorded cash receipts were traced into the depositories; the cancelled bank vouchers were inspected and compared with the cash records and the dues income was accounted for.

Submitted herewith is a Balance Sheet showing the financial condition of the Society on December 31, 1939, and your attention is directed to the following comments thereon:

CASH

Cash on Deposit was verified by direct communication with the banks and reconcilement of the amounts reported to us with the balance shown by the books of the Society.

The Petty Cash kept in the New York Office was vertified by count.

MARKETABLE SECURITIES

There is attached hereto a schedule of negotiable bonds which were verified by direct communication with the Bankers Trust Company where same are deposited for safe-keeping. No adjustment has been made of the $3,280.21 shrinkage in the market value of these securities, same being included in the attached Balance Sheet at cost.

ACCOUNTS RECEIVABLE

A list of the Membership Dues Receivable as of December 31, 1939, furnished to us by the management was checked to the individual ledger cards and found in agreement with the General Ledger Control. The unpaid dues may be summarized as follows:

Dues billed in 1939	$ 9,861.46
Dues billed in 1938	2,961.59
Dues billed in prior years	4,643.44
TOTAL	$17,466.49

Amounts due from GUIDE Advertisers and other Accounts Receivable were verified by trial balance of the individual ledger; accounts were found in agreement with the General Ledger Control.

After charging off $3,939.44 in uncollectible dues per council authorization we have increased the Reserve for Dues Doubtful of Collection $7,630.28. The adjusted reserves now shown on the attached Balance Sheet in our opinion are ample to cover any losses from future realization.

INVENTORIES

Items appearing under this caption are based upon quantities submitted by the management and computations made by us. The Transactions inventory priced at cost may be summarized as follows:

Prior Years	$1,750.00
1933–1934	670.46
1935–1936	75.90
1937–1938	448.72
TOTAL	$2,945.08

FURNITURE, FIXTURES AND LIBRARY

Furniture, Fixtures and Library are shown herein at the book values without appraisal by us. We did, however, provide for Depreciation of Furniture and Fixtures at the rate of ten per cent (10%) per annum.

ACCOUNTS PAYABLE

On December 31, 1939, there remained unpaid invoices amounting to $9,976.00 which included the sum of $9,407.42 estimated by the management necessary to complete and make the first mailing of the 1940 GUIDE.

PAYABLE TO RESEARCH FUND

The amounts due the Research Fund, representing 40% of Members' and Associates' dues, were determined from computations made by us in accordance with Section 5, Article 3, of the By-Laws.

ACCRUED ACCOUNTS

Additional compensation to the secretary and the clerical staff of the Society has been computed in accordance with the instructions of the Finance Committee.

DEFERRED INCOME

Members' prepaid dues were ascertained by trial balance of the individual ledger cards. Prepaid dues from proposed members were verified from inspection of the applications found on file.

GENERAL FUND

There is attached hereto a schedule showing the changes that have occurred in the General Fund during the calendar year 1939.

SPECIFIC FUNDS

Specific Funds are shown in the attached Balance Sheet after adjustment for interest earned.

The Balance Sheet and schedules showing the operations of the Research fund are annexed hereto, also a separate report has been prepared and rendered to the Committee on Research.

OPERATIONS

There are attached hereto Statements of Income and Expenses for the calendar year 1939 showing an excess of Expenses Over Income from Society activities of $3,959.52 and a net income of $5,602.39 from GUIDE Operations. In the preparation of these statements 30% of the salaries and office expenses have been allocated to GUIDE Operations.

Respectfully submitted,
TUSA & LA BELLA,
Certified Public Accountants.

BALANCE SHEET

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
NEW YORK, N. Y.

(December 31, 1939)

ASSETS

SOCIETY

CASH

On Deposit	$36,675.12	
On Hand	4.03	
In Closed Banks	418.97	$ 37,098.12

RESERVE FUNDS

On Deposit		$ 4,294.29

INVESTMENTS (AT COST)

Securities (Market Value $18,919.17)	$18,678.23	
Add: Accrued Interest	4.50	$ 18,682.73

ACCOUNTS RECEIVABLE

Membership Dues	$17,466.49		
Less: Reserve for Doubtful	11,796.69	$ 5,669.80	
Advertisers and Sundry Debtors	$31,098.81		
Less: Reserve for Doubtful	1,462.62	$29,636.19	$ 35,305.99

INVENTORIES

Transactions—Copies	$ 2,945.08	
Transactions—Paper	371.28	
Emblems and Certificate Frames	183.35	
Stationery	116.38	$ 3,616.09

PERMANENT

Library		$300.00	
Furniture and Fixtures	$ 7,209.69		
Less: Reserve for Depreciation	5,447.56	1,762.13	2,062.13
			$101,059.35

SPECIFIC FUNDS

ENDOWMENT FUND

Cash on Deposit		$13,828.19	
Securities at Cost (Market Value $8,397.50)	$11,918.65		
Add: Accrued Interest	111.35	$12,030.00	$ 25,858.19

F. PAUL ANDERSON AWARD FUND

Cash on Deposit		1,126.01
		$128,043.55

LIABILITIES AND GENERAL FUND

SOCIETY

ACCOUNTS PAYABLE... $ 9,976.00

PAYABLE TO RESEARCH FUND
On Dues as and when Collected.................................... 6,342.61

ACCRUED ACCOUNTS
Compensation—Secretary and Staff................................ 2,614.21

RESERVE FOR PUBLICATION
Transactions—Volume 45... 3,700.00

DEFERRED INCOME
Prepaid Dues—Members................................ $ 346.42
Prepaid Dues—Prepaid Members........................ 316.63 $ 663.05

TOTAL LIABILITIES... $ 23,295.87

GENERAL FUND
Society... $ 77,763.48

TOTAL LIABILITIES AND GENERAL FUND....................... $101,059.35

SPECIFIC FUNDS

ENDOWMENT FUND
Principal... $24,063.28
Unexpired Income.................................... 1,794.91 $ 25,858.19

F. PAUL ANDERSON AWARD FUND
Principal... $ 1,000.00
Unexpired Income.................................... 126.01 $ 1,126.01

$128,043.55

NOTE "A"—This Balance Sheet is subject to the comments contained in the letter attached to and forming a part of this report.

BUDGET COMPARISON

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
NEW YORK, N. Y.

(For the Calendar Year Ended December 31, 1939)

	Actual	Budget Provision	Increases Decreases
INCOME			
Initiation Fees...	$ 2,547.00	$ 1,500.00	$1,047.00
Dues Collectible..	24,318.09	26,000.00	1,681.91
Editorial Contract..	15,648.31	15,000.00	648.31
Profit—Emblems and Certificate Frames.......................	98.12	100.00	1.88
Profit—Reprints and Books....................................	232.55	300.00	532.55
Sales of TRANSACTIONS..	1,077.73	750.00	327.73
Interest—Securities and Savings Accounts.....................	1,213.93	1,000.00	213.93
GUIDE Advertising..	27,440.10	27,000.00	440.10
GUIDE Sales..	18,966.97	26,000.00	7,033.03
	$91,077.70	$97,650.00	$6,572.30
EXPENSES			
General Society Activities			
Meetings..	$ 2,727.77	$ 1,800.00	$ 927.77
Chapter Allowances..	677.94	1,000.00	322.06
Promotional Expense...	589.71	1,000.00	410.29
Speakers Bureau...	1,209.02	1,500.00	290.98
Subscriptions—HPAC...	6,308.23	6,300.00	8.23
Postage...	1,712.19	2,000.00	287.81
Rent and Light..	3,601.99	4,000.00	398.01
TRANSACTIONS...	4,467.79	3,700.00	767.79
Codes..		1,000.00	1,000.00
Salaries—Secretary and Staff................................	17,731.00	18,100.00	369.00
General Printing..	499.35	700.00	200.65
Membership Certificates and Emblems........................	185.10	600.00	414.90
President's Fund..	1,100.20	1,800.00	699.80
Secretary Travel..	1,450.03	1,200.00	250.03
Council and Chapter Delegates Travel........................	3,413.75	4,000.00	586.25
Bank Charges...	103.36	100.00	3.36
Multigraphing..	280.63	600.00	319.37
Telephone..	661.94	700.00	38.06

Telegraph	$ 275.30	$ 350.00	$ 74.70
Professional Services	500.00	650.00	150.00
General Office Expenses	555.58	650.00	94.42
Office Supplies	659.49	600.00	59.49
Furniture and Fixtures	240.97	500.00	259.03
Reserve for Additional Compensation	2,614.21	2,100.00	514.21
Special Appropriations	5,268.12	1,200.00	4,068.12
	$56,833.67	$56,150.00	$ 683.67

COST OF GUIDE

Copy Sales Promotion (1939 Edition)	$ 3,285.90	$ 3,500.00	$ 214.10
Expressage and Mailing (1939 Edition)	3,028.54	4,500.00	1,471.46
Advertising Sales Promotion	1,452.32	3,000.00	1,547.68
Editorial and Advertising Salaries	7,000.00	7,500.00	500.00
Committee Expense	441.89	500.00	58.11
Printing 1940 Edition, 13M Copies	9,008.62	12,000.00	2,991.38
Binding 1940 Edition	4,013.25	4,500.00	486.75
Paper 1940 Edition	2,879.08	4,000.00	1,120.92
Cuts and Drawings	429.27	500.00	70.73
Yearbook	1,062.29	1,500.00	437.71
	$32,601.16	$41,500.00	$8,898.84
	$89,434.83	$97,650.00	$8,215.17

BALANCE SHEET

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
RESEARCH FUND—NEW YORK, N. Y.

(December 31, 1939)

ASSETS

CASH

ON DEPOSIT

Treasurer's Account—Bankers Trust Co	$2,013.28	
Secretary's Account—Chase National Bank	1,205.53	
Director's Account—Forbes National Bank	77.96	
Thrift Account—Bank for Savings	3,148.39	$6,445.16

ON HAND FOR DEPOSIT

Treasurer's Account—Bankers Trust Co	3,050.60	
Director's Account—Forbes National Bank	224.95	3,275.55

ON HAND

Petty Cash—Pittsburgh	16.29	$ 9,737.00

ACCOUNTS RECEIVABLE

RESEARCH CONTRACTS

Navy Department	2,081.09	
Pittsburgh Corning Corp	250.00	
Owens Illinois Glass Co	250.00	2,581.09

PERMANENT

Laboratory Equipment	1.00

Total Research Fund Assets	$12,319.09

FUND

RESEARCH ENDOWMENT FUND

CASH ON DEPOSIT

Bank for Savings	697.93
	$13,017.02

LIABILITIES AND FUND

ACCOUNTS PAYABLE

Trade Creditors	$ 326.40

DEFERRED INCOME

Radiation and Comfort Study	$2,017.69	
Air Conditioning Requirements of Glass	846.83	
Corrosion Study	500.00	
Convector Study	400.00	
Air Distribution	750.00	4,514.52 $4,840.92

RESEARCH FUND.................................... $7,478.17

    Total Research Fund Liabilities and Fund....................... $12,319.09
FUND
  RESEARCH ENDOWMENT FUND
    Principal.. 600.00
    Unexpired Income................................. 97.93   697.93

                                   $13,017.02

NOTE "A"—This Balance Sheet is subject to the comments contained in the letter attached to and forming a part of this report.
NOTE "B"—The Research Fund as of December 31, 1939, had a contingent asset amounting to $6,342.-61 arising from 40% of the Society's members and associates dues, payable upon collection by the latter.

BUDGET COMPARISON—RESEARCH FUND

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
NEW YORK, N. Y.

(For the Calendar Year Ended December 31, 1939)

	Actual	Budget Provision	Increases Decreases*
Balance from 1938 Committee—Cash in Pittsburgh and New York Banks..	"A"$15,253.55	$15,253.55	$15,253.55*
INCOME			
BUDGETED			
Dues—Current and Prior Years........................	$14,903.73	16,000.00	1,096.27
Special Council Appropriations.........................	5,000.00	5,000.00	....
Interest on Savings Accounts..........................	62.17	60.00	2.17
Contributions—General Research.......................	2,425.00	2,000.00	425.00
Contributions—Earmarked Research—Radiation and Comfort..	3,982.31	6,000.00	2,017.69
Contributions—Earmarked—Glass Study...............	1,562.80 "A"	2,409.63	846.83
	$27,936.01	$46,723.18	$18,787.17
UNBUDGETED			
Earmarked Research—Navy Contract...................	$ 5,000.00	....	$ 5,000.00
Earmarked Research—Glass Block......................	3,500.00	....	3,500.00
Earmarked Research—Cooling Load....................	200.00	....	200.00
	$36,636.01	$46,723.18	$10,087.17
EXPENDITURE			
BUDGETED			
Salaries—Laboratory Staff.............................	$ 9,419.60	$13,500.00	$ 4,080.40
Salaries—Student Help................................	477.50	1,800.00	1,322.50
Salaries—Correlation Thermal Research.................	1,500.00	1,500.00	....
Salaries—Office of Chairman...........................	300.00	300.00	....
Travel—Committee and Laboratory Staff................	1,647.93	3,000.00	1,352.97
Laboratory Supplies and Equipment....................	873.98	1,000.00	126.02
Office Supplies and Equipment.........................	781.47	600.00	181.47
Printing..	159.39	250.00	90.61
Meetings...	213.16	300.00	86.84
Exhibits..	226.40	250.00	23.60
Promotion and Publicity...............................	222.67	500.00	277.33
Contract—U. S. Bureau of Mines.......................	101.83	500.00	398.17
Cooperative Research..................................	8,525.00	10,550.00	2,025.00
Earmarked Research—Radiation and Comfort...........	3,982.31	6,000.00	2,017.69
Earmarked Research—Glass Study.....................	1,562.80	2,409.63	846.83
Earmarked Research—Cooling Load....................	4,345.74	600.00	3,745.74
Earmarked Research—Air Conditioning.................	1,147.25	2,500.00	1,352.75
Contingencies..	165.47	500.00	334.53
	$35,651.60	$46,059.63	$10,408.03
UNBUDGETED			
Earmarked Research—Navy Contract...................	$ 5,000.00	....	$ 5,000.00
Earmarked Research—Glass Block......................	3,610.79	....	3,610.79
	$44,262.39	$46,059.63	$ 1,797.24
Balance..	7,626.38	663.55	8,289.93
	$36,636.01	$46,723.18	$10,087.17

The report of the Council was given by the secretary, as follows:

Report of Council

To administer the business of the Society the Council held five meetings during the year just completed. The organization meeting was held January 26, 1939 at Pittsburgh, Pa., with President McIntire presiding. He announced the appointment of five Council Committees and five Special Committees, and the Chairman and Vice-Chairman of the Committee on Research respectively, Messrs. W. L. Fleisher and J. H. Walker.

Additional special committees were appointed during the year as needed.

Appointment of the Secretary and Technical Secretary was made and depositories for Society funds were selected. The Budget for 1939 provided for an estimated income of $97,650.00 and an expenditure of $97,650.00. The Research Budget amounting to $27,936.00 was approved, and $5,000 was appropriated from Society funds for research work.

The accounting firm of Tusa and La Bella, Certified Public Accountants, was selected to audit Society accounts and rendered two reports during the year.

During the summer, two meetings of the Council were held at Mackinac Island, July 4 and 6, at which reports of the Special Committees were received. Approval was given to the investment of $7,500 in United States Savings Bonds (legal limit permitted by law).

A Charter was granted for the organization of a Local Chapter in Portland, Ore., upon petition of 36 members. Nominees for a three-year term on the Committee on Research commencing in 1940 were made.

The Council inaugurated a new plan and authorized a fall meeting of the Society in Atlanta, Ga., October 30 and 31.

Washington, D. C. was selected as the meeting place for the summer meeting 1940.

The fourth meeting of the Council in 1939 was held in Atlanta, Ga., October 29, and reports were received from regular and special committees. Provision was made for payment of chapter delegates' transportation expenses to the 46th Annual Meeting.

A Committee was appointed to review the Society's publication policy and negotiated a new editorial contract. The Exchange Service Plan with the *ASRE* was approved for 1940.

Legal advice was sought on the status of members who had given notes and agreements for deferred payment of Society's dues several years ago.

During the year routine action was taken in providing for election of members to the Society, acceptance of resignations, cancellations as required by the Constitution and By-Laws.

Petition for the 29th Chapter of the Society from New Orleans members was received and voted upon by letter ballot and a Charter was granted for the Delta Chapter of the ASHVE with headquarters in New Orleans.

On January 22 the final meeting of the Council was held in Cleveland, Ohio, and received reports of committees and other unfinished business.

Respectfully submitted,
A. V. HUTCHINSON, *Secretary.*

President McIntire then presented the Report of the Tellers as follows:

Report of Tellers of Election

BALLOT FOR OFFICERS

	For
President—F. E. Giesecke	634
First Vice-President—W. L. Fleisher	619
Second Vice-President—E. O. Eastwood	633
Treasurer—M. F. Blankin	633

Members of the Council—Three-Year Term

J. F. Collins, Jr. 634
E. N. McDonnell. 633
T. H. Urdahl. 634
C.-E. A. Winslow. 634
   Total Ballots Counted. 698
   Ballots Invalid. 64

BALLOT FOR COMMITTEE ON RESEARCH

Three-Year Term

Philip Drinker. 633
Axel Marin. 633
A. E. Stacey, Jr. 634
J. H. Van Alsburg. 630
J. H. Walker. 634
   Total Ballots Counted. 698
   Ballots Invalid. 64

BOARD OF TELLERS
C. H. QUIRK, *Chairman*
W. W. TIMMIS
H. S. WHELLER

L. T. Avery, chairman of the Committee on Arrangements for the Northern Ohio Chapter was introduced and welcomed the members and guests to Cleveland on behalf of the Chapter, and outlined some of the entertainment features.

President McIntire called the third session to order on Wednesday morning, January 24, and announced the Report of the Committee on Research presented by W. L. Fleisher, chairman of Committee on Research, reported as follows:

Report of Committee on Research

The Committee on Research closed its 1939 activity after an eminently successful year. Three meetings of the entire Committee were held, two during the Annual Meeting in January and one during the Semi-Annual Meeting in July. At these meetings the Committee organized, prepared and directed the operations under its budget and reviewed, and in many respects guided, the work of its 24 sub-committees. These committees carried on their activities through frequent meetings and correspondence and as a result have been very effective in analyzing the research needs in their respective lines which have, in many cases, resulted in the outline of definite research projects. Where it was possible for such projects to be initiated the Technical Advisory Committee cooperated closely with the Director of the Laboratory and cooperating institutions conducting the work.

Of particular interest during the past year has been the consummation of an agreement with the U. S. Navy for conducting a comprehensive cooperative program, and the Executive Committee and Director of the Laboratory have given unstintingly of their time to further the interests of the Government.

A study of radiation as a factor in comfort was another new project initiated. This question has been on the Committee's agenda since the publication of the first work on effective temperature. Through the diligent persistence of the entire Committee sufficient special funds were obtained to build and equip, with new and much needed instruments, three structures at the Pittsburgh Laboratory for the investigation of authorized research projects. These structures are now the property of the Society and may be used for other research work when the present programs for which they were built are completed.

The vast scope of Society research and its accomplishments in the past year under the several Technical Advisory Committees follows. ·

Reports of Technical Advisory Committees

SENSATIONS OF COMFORT, 1a—C. Tasker, *Chairman;* C. R. Bellamy, Thomas Chester, Elliott Harrington, W. S. Kilpatrick, Dr. W. J. McConnell, A. B. Newton, B. F. Raber, L. C. Soule.

Data which may be considered as a summary of a number of field tests which have been conducted on summer cooling requirements in various localities were presented in the form of a paper at the 1939 Semi-Annual Meeting of the Society. Conclusions based on these tests would seem to indicate that localities having a maximum normal mean summer temperature of 74 F or above should maintain an effective temperature of about 71 deg, while those localities showing normal mean summer temperatures of below 74 F should maintain effective temperatures below 71 deg. Beyond this, some adjustment, probably not to exceed one degree effective temperature, might be based upon peculiar local conditions or peculiarities of the type of occupant.

Investigations are planned for 1940 at the Society Research Laboratory to determine the reactions of subjects to an optimum effective temperature at low relative humidities, and the relation of these reactions to those experienced at the same effective temperatures at high humidities, both as regards feeling of warmth and any other sensations which may be affected.

PHYSIOLOGICAL REACTIONS, 1b—C.-E. A. Winslow, *Chairman;* Dr. Thomas Bedford, Dr. E. F. DuBois, Dr. R. W. Keeton, André Missenard, Dr. R. R. Sayers, Dr. Charles Sheard, C. Tasker.

It is considered the task of this Committee to present at intervals summaries of current physiological researches and their bearing on ventilation practice. A fairly comprehensive report of this Committee was presented at the 1939 Annual Meeting, and it did not seem advisable to review the subject again during the present year. By 1941 it should probably be possible for the Committee to present another report.

Through a cooperative agreement with the Medical School at the University of Illinois, Chicago, important research studies have been continued on the peripheral type of circulatory failure in experimental heat exhaustion as related to the role of posture of an individual, detailed results of which will be presented at the 1940 Annual Meeting.

TREATMENT OF DISEASE, 1c—Dr. M. B. Ferderber, *Chairman;* Dr. C. J. Barone, Dr. A. R. Behnke, Dr. B. Z. Cashman, W. L. Fleisher, Dr. T. Lyle Hazlett, C. S. Leopold, Dr. C. D. Selby, Dr. A. W. Sherrill, Dr. H. F. Smith, Dr. C. S. Stephenson, Dr. B. L. Vosburgh.

Through a cooperative agreement with the School of Medicine, University of Pittsburgh, work has been continued at the Magee Hospital, to study the application of air conditions to operating rooms and recovery wards. Investigations this year dealt solely with the bacteriological content of the air in the operating rooms.

Equipment is available at the hospital to clean the air with either a cloth filter, electrostatic precipitator, or air washer. Research is to determine whether these various systems of air cleaning or purification actually decrease the bacteriological

content of the air delivered to an operating room. Two methods of taking the samples are being used, first, involving blood agar pour plates to determine the amount of settling bacteria, and the second involves air samples taken by a centrifuge method. An ultraviolet sterile lamp is also being used as a method of bacterial control of the air within the operating room.

This Committee has also concerned itself with the wider application of the use of fever therapy in treating various diseases. Recent reports have indicated that this method of treatment is being used in industrial hospitals with encouraging results.

CLIMATE AND SEASON, *1d*—J. H. Walker, *Chairman;* Dr. H. A. Abramson, O. W. Armspach, Ellsworth Huntington, Dr. C. A. Mills, André Missenard, T. H. Urdahl.

The general objective of this Committee is to study the effect of climate and season on the physiological reactions of human beings, particularly as to their relation to the air conditioning problem. A bibliography has been compiled, which is in the files of the Bureau for Correlating Thermal Research, at the Research Laboratory. Copies may be obtained upon request.

AIR CONDITIONING IN INDUSTRY, *1e*—A. E. Stacey, Jr., *Chairman;* Philip Drinker, Dr. Leonard Greenburg, H. P. Greenwald, A. M. Kinney, J. W. Kreuttner, L. L. Lewis, Dr. W. J. McConnell, Dr. C. P. McCord, P. A. McKittrick, Dr. R. R. Sayers, Dr. Charles Sheard, C. Tasker, R. M. Watt, Jr.

The program of this Committee for the current year called for determination of the effect of acclimatization to atmospheric environments for different seasons of the year. During the winter months of February and March, some data were collected at the Society's Research Laboratory which showed conclusively that a person gets a greater rise in body temperature and increase in pulse rate due to exposure under the same hot atmospheric environment than is realized during the summer. The shift to a lower effective temperature approximates that between the summer and winter comfort zone, or about 5 deg. The leucocyte count data of February and March might indicate an interesting effect of environmental conditions. The data collected during April show some change; however, not as much as in the case of February and March. A small amount of data were obtained during August to verify the results of the previous summer.

Only a very small amount of work has been done to determine, if possible, any difference in physiological reactions of persons in the same atmospheric environments, but with high and low moisture content. From one test, no material variation in rise of body temperature, or increase in pulse rate was noted. This is the next work on the program of the Committee, as there seems to be some interesting results in connection with the leucocyte count.

Two other points of investigation are being considered: (1) to obtain data on the relative physiological changes of men and women in exposure to hot environments; and (2) to obtain physiological reactions of persons at different rates of activity to atmospheric environments.

AIR POLLUTION AND PURIFICATION, *2*—C.-E. A. Winslow, *General Chairman.*

AIR POLLUTION, *2a*—H. B. Meller, *Chairman;* F. A. Chambers, Philip Drinker, J. S. Owens, Sol Pincus, H. J. Rose, R. J. Tenkonohy, E. C. Webb, E. H. Whitlock.

REMOVAL ATMOSPHERIC IMPURITIES, *2b*—Dr. Leonard Greenburg, *Chairman;* R. D. Bennett, R. S. Dill, Theodore Hatch, L. R. Koller, F. H. Munkelt, G. W. Penney, Dr. E. B. Phelps, F. B. Rowley, C. Tasker, W. O. Vedder, Jack Waggoner, W. F. Wells, Dr. Renée Eulenburg Wiener.

A great deal of attention has been devoted during the year to the reorganization of this committee. Two excellent groups have been established to deal with the general problems of air pollution and removal of atmospheric impurities. A general meeting of this group was held in New York on December 14 and as an initial program steps were taken to formulate a comprehensive report showing the present status of all knowledge now available covering this subject.

RADIATION AND COMFORT, 3—J. C. Fitts, *Chairman*; A. H. Barker, L. M. K. Boelter, R. E. Daly, E. R. Gurney, L. N. Hunter, C. S. Leopold, A. P. Kratz, D. W. Nelson, G. W. Penney, W. R. Rhoton, C.-E. A. Winslow.

This Committee was appointed in 1938 to consider all problems relating to radiation as it affects comfort in winter and summer. A comprehensive outline of basic researches was prepared and it was decided to concentrate first on a study to determine the primary sense reactions of relative comfort derived from rooms with normal window arrangements from convected heat and from heat supplied by both convection and direct radiation.

During the summer of 1939 a building was erected at the Society's Research Laboratory in Pittsburgh consisting of two rooms with identical exposures. These rooms have been equipped with the essential apparatus to supply heat and the necessary controlling and measuring devices so that they can be used to carry forward the investigations originally projected by this Committee.

Up to the time of preparing this report only preliminary tests were made for the purpose of observing the equipment. It is the plan of the Committee to carry forward its program of tests as soon as weather conditions in Pittsburgh give consistently cold weather during the daylight hours.

INSTRUMENTS, 4—D. W. Nelson, *Chairman*; F. R. Bichowsky, L. M. K. Boelter, R. S. Dill, M. K. Fahnestock, A. P. Gagge, J. A. Goff, G. L. Tuve, C. P. Yaglou.

This Committee was formed during the early part of 1939 to function as a coordinating group and to study all phases of research dealing with measurements. Due to the great amount of research work now under investigation in various laboratories, it is recognized that a uniform method of measuring thermal environment is desirable and attempts toward standardization in this field is an objective of this group.

Several research projects now being carried on in cooperative institutions dealing with air distribution and air friction require accurate means of calibrating air velocity instruments, and this group is now actively assisting in order that the measurements being taken at various laboratories may be interpreted on a uniform basis. The Committee is also encouraging the development of new instruments applicable to the heating, ventilating and air conditioning industry and in this connection papers presented before the Society in 1939 have indicated the development of several interesting instruments such as a thermocouple psychrometer, a pressure plate anemometer, a radiometer and an absolute hygrometer.

WEATHER DESIGN CONDITIONS, 5—T. H. Urdahl, *Chairman*; J. C. Albright, John Everetts, Jr., E. W. Goodwin, J. B. Kincer, O. A. Kinzer, J. W. O'Neill, L. S. Ourusoff, F. W. Reichelderfer.

During the past year the activities of this Committee have been devoted to furthering a means for continuation of the work begun in 1938, especially in determining climatic differences existing between Weather Bureau Stations and locations in the center of a city, which are subject to different influences.

The Committee held two meetings in Washington, with officials of the Weather Bureau, and tentative plans have been made for a cooperative research project between the Society and the U. S. Weather Bureau, in establishing a few stations to determine the existing temperature differences between regular stations and city locations.

TRANSPORTATION AIR CONDITIONING, 6—A. E. Stacey, Jr., *Chairman*; W. I. Cantley, T. R. Crowder, A. G. Dixon, C. C. Elmes, L. H. Laffoley, E. A. Russell, J. H. Van Alsburg, W. E. Zieber.

The Committee has continued this year for the purpose of correlating and consolidating existing material. No special research project was outlined for detailed consideration as it was felt that the main purpose of this group should be to act as a clearing house for the dissemination of data suitable for ship, bus, automobile, aeroplane, and railroad application. In the event that essential research is required the committee feels that it will be its function to suggest the work to be done to fill in present gaps of knowledge.

RADIATION WITH GRAVITY AIR CIRCULATION, 7—M. K. Fahnestock, *Chairman;* R. E. Daly, A. G. Dixon, H. F. Hutzel, J. P. Magos, J. W. McElgin, J. F. McIntire, T. A. Novotney, W. A. Rowe.

Under this Committee the work during 1939 has been a continuation of a program in progress for a number of years as a cooperative project with the University of Illinois. The work may be divided into two parts, *first,* that which is being conducted in a warm wall testing booth and, *second,* that which is being conducted in a room heating test plant.

Under the first program standardization tests with steam as a heating medium were completed this year on two light weight cast-iron radiators and on two convectors with cast-iron heating units. These tests were part of a program for standardizing and improving test methods and eventually these same units are to be tested in several different laboratories and the results tabulated and studied.

Under the second study the performance of conventional tubular steam radiators and steam convectors is being studied in a completely remodelled room heating test plant. Three types of direct cast-iron radiators and three types of convectors are being used in these tests with two walls of the test room exposed. It is expected that the results will indicate the effect of size, shape and exposure of the test room upon the performance of tubular steam radiators and convectors and should confirm the correctness of adopted test methods or afford information for improving them.

HEAT TRANSFER OF FINNED TUBES WITH FORCED AIR CIRCULATION, 8—G. L. Tuve, *Chairman;* William Goodman, W. E. Heibel, H. F. Hutzel, S. F. Nicoll, R. H. Norris, L. P. Saunders, C. F. Wood.

Since the Committee felt that its most important function was correlation and consolidation of existing material, its efforts have been directed toward obtaining an agreement on terminology, fundamental theory, and experimental methods. During the first part of the year an experimental study was carried on at Case School of Applied Science, in which several of the calculation methods for dehumidifying coils were applied to the same sets of test data. These test data were carefully recorded by precision methods and the final result was a surprising agreement between the various methods of computation. Recommendations for a Code for Testing and Rating Fin-Tube Coils have been discussed at some length by the Committee. A memorandum on definitions and one on test methods has been issued for comment.

COOLING LOAD IN SUMMER AIR CONDITIONING, 9—C. S. Leopold, *Chairman;* C. M. Ashley, John Everetts, Jr., F. H. Faust, M. G. Kershaw, A. E. Knapp, L. S. Morse, A. E. Stacey, Jr., R. M. Stikeleather, J. H. Walker.

The activity of this Committee during the past year included the development of plans for a study of the effect of sun radiation on heat gain through building walls. The Research Laboratory built a cubicle having inside dimensions of 18 x 18 ft by approximately 8 ft high. The east, south and west side walls of this building each contained panels of 13-in. brick, 4-in. brick, 8-in. hollow tile, and 4-in. brick veneer on frame construction. The northern exposure differed in that the frame veneer of the construction was eliminated in order to make room for a door.

The roof provided for nine horizontal roof panels all covered with felt and tar waterproofing. These panels included three 2-in. concrete, one 6-in. concrete, two 2-in. plank, one 3-in. tile, one 2-in. gypsum, and one 4-in. gypsum. Provision was made for flooding one of the 2-in. concrete and one of the 2-in. wooden panels. These could be either sprinkled to just keep them wet or flooded with water up to a depth of 6 in. One of the remaining 2-in. concrete panels was finished with the black tar exposed while the others were covered with slag.

The cubicle interior was provided with a cooling facility which with some modification can be converted into a heating system for winter study. Tests were made on several days during part of July and August in which the rates of heat transfer through the inside surfaces of the eleven wall sections and nine roof panels were determined through the 24-hour cycle with the Nicholls' heat flow meter while simultaneous temperatures of the outside surface, inside surface, the outside air, and the inside air and on a number of intermediate points were taken. A great mass of data was accumulated as a result of the summer study giving the simultaneous heat gain through the walls of different construction and exposure.

Solid Fuels, *10*—R. A. Sherman, *Chairman;* W. A. Danielson, R. S. Dill, H. N. Eavenson, A. C. Fieldner, L. N. Hunter, A. J. Johnson, R. E. Kerr, H. K. Kugel, P. Nicholls, V. F. Parry, H. J. Rose, D. M. Rugg, J. E. Schoen, L. E. Seeley, E. T. Selig, R. T. Smith, C. Tasker.

In the first full calendar year since its organization in May 1938, this Committee has actively pursued its functions, namely, the dissemination of available knowledge on equipment and methods for the utilization of solid fuels, the encouragement of research to obtain further knowledge, and investigations to determine the need for codes for testing, and performance of solid-fuel-burning equipment. The work was carried on by extensive correspondence and by three meetings of the Committee, held January 25 in Pittsburgh, April 27 in Primos, Pa., and October 6 in Columbus, Ohio.

The Committee sponsored two research projects during the year. One of these dealt with the performance of domestic chimneys under varying conditions. The second project, in two parts, is being carried on at the Experiment Station of the U. S. Bureau of Mines, Golden, Colorado. One phase of this investigation will include a study of the performance of sub-bituminous coals with residential under-feed stokers, and the second phase of the study covers the performance of these coals in a stoker-fired central heating plant on the campus of the Colorado School of Mines.

Working committees have been appointed by the sub-committees on anthracite, bituminous, and sub-bituminous coal and lignite, to collect information on various types of solid-fuel-burning equipment. Also a special sub-committee has been appointed to investigate the need for codes for testing and performance of water heaters fired with solid fuels.

Summer Air Conditioning for Residences, *11*—M. K. Fahnestock, *Chairman;* C. F. Boester, E. A. Brandt, John Everetts, Jr., Elliott Harrington, H. F. Hutzel, J. A. Kiesling, E. D. Milener, K. W. Miller, R. E. Robillard, F. G. Sedgwick, J. H. Walker.

No active research work was conducted under the auspices of this Committee during 1939. The Committee was continued for the purpose of sponsoring at least one summer's cooling studies in the Research Residence after it was completely insulated in order to compare results from an insulated residence with those from an uninsulated residence. During the summer of 1939 the Research Residence was completely insulated and pending the acceptance of a satisfactory program it is contemplated that a series of tests will be conducted during the summer of 1940 similar to those which were made in 1934 in the uninsulated residence. In that year mechanical refrigeration was used for cooling and both the first and second floors were cooled. At the present time there is some indication that a plan for research in gas summer air conditioning will be initiated in the near future.

Air Distribution and Air Friction, *12*—J. H. Van Alsburg, *Chairman;* S. H. Downs, M. K. Fahnestock, R. D. Madison, Axel Marin, L. G. Miller, D. W. Nelson, C. H. Randolph, D. J. Stewart, Ernest Szekely, M. C. Stuart, R. J. Tenkonohy, G. L. Tuve.

Work outlined by this Committee has developed new data this year based on research being conducted at the Society Laboratory and in four cooperative institutions. A paper presented before the Society in January, 1939 covered the frictional resistance to the flow of air in straight, round and square ducts. Investigations this year have dealt primarily with air flow resistance in rectangular ducts and elbows. All of this program is aimed to complete the Guide tables on duct friction and because of its basic importance will be of tremendous value to the industry.

Cooperative work was initiated at Lehigh University this year to determine the friction of elbows with particular emphasis on the effect of turning vanes and blades. The program at Case School of Applied Science has been planned to determine data on instrument calibration used in air friction and air distribution research and also to determine the nature of the spread of air streams from discharge outlets and the entrainment of room air by such streams. A paper covering stackhead performance is being presented at the Annual Meeting covering cooperative research at the University of Wisconsin during the current year.

The cooperative investigation on air distribution being conducted at the University of Illinois is primarily concerned with the effect of inlet and exhaust opening location

on the motion and distribution of air within a room. The tests are being conducted in an experimental room located within a larger insulated enclosure provided with direct expansion refrigeration coils and electric heaters so that control temperatures from 0 to 110 F may be maintained around the outside of the experimental room. Present plans are to determine air distribution with various inlet air velocities ranging from 300 to 1200 fpm when cooling the room under varying conditions.

HEAT REQUIREMENTS OF BUILDINGS, *13*—P. D. Close, *Chairman;* W. H. Badgett, E. K. Campbell, J. F. S. Collins, Jr., E. F. Dawson, W. H. Driscoll, H. M. Hart, H. H. Mather, F. B. Rowley, J. H. Walker.

Among the problems considered by this committee during 1939 was that relating to the loss of heat through basement floors and walls. In arriving at the heat transmission coefficients for concrete basement floors on the ground, it has been customary to consider only the concrete floor and any other materials in the floor construction, and to neglect any possible heat-resisting effect of any part of the ground on which the floor is located.

It is probable that calculations based on this premise, especially in the case of heated basements, will result in excessive heat losses. However, there appears to be no valid reason that the heat resistance of the floor stops at the under surface of the concrete. On the contrary, it is axiomatic that the heat transfer takes place from a point above the basement floor to a point in the dirt a certain distance below the basement surface where the temperature gradient stops. The intervening dirt will therefore have some heat resistance value. The termination of this temperature gradient will depend on a number of factors, including the rate at which heat is dissipated to the ground, which in turn, depends on the rate of heat transfer of the ground.

A study of this subject is under consideration for both basement floors and walls below grade and with and without insulation. It is expected that the results of this study will make it possible to check, or perhaps revise, the Guide data on basement heat losses.

Another problem which has been under consideration for some time is that dealing with the selection of temperatures and wind velocities for calculating heat losses. For maximum accuracy, concurring combinations of temperatures and wind velocities should be selected rather than combinations which have no relation to each other. However the solution of this problem is complicated by the fact that the concurring construction which will give the maximum heat loss will depend largely upon the type of structure, especially the relation between the transmission and infiltration losses, because for example, a high wind velocity will not have the same relative effect upon these heat losses in all cases. The problem is further complicated by the fact that the results will vary in each locality. Some progress is being made, however, and it is expected that a satisfactory solution of the problem will be reached at an early date.

AIR CONDITIONING REQUIREMENTS OF GLASS, *14*—M. L. Carr, *Chairman;* F. L. Bishop, W. A. Danielson, H. C. Dickinson, J. E. Frazier, S. O. Hall, E. H. Hobbie, C. L. Kribs, Jr., R. A. Miller, F. W. Parkinson, J. H. Plummer, W. C. Randall, L. T. Sherwood, J. T. Staples, G. B. Watkins, F. C. Weinert.

Two meetings of the Committee were held during the past year, one during the Annual Meeting and the other on October 19, in Pittsburgh. The Committee has continued to maintain its own budget and to employ a research engineer at the Society's Research Laboratory. During the past summer the Research Laboratory, in cooperation with the Committee conducted a study on solar heat gain through glass blocks and a paper resulting from this study will be presented at the 1940 Annual Meeting. In conjunction with this study, another investigation on the determination of air to air transmittance values for glass blocks was carried out at the Pittsburgh Testing Laboratory under the supervision of the Committee and the Research Laboratory. A paper resulting from this work will be published in the near future.

The Committee is planning to conduct an extensive research program on the heat transmitting properties of flat glass in single and double glazed units of steel and wood sash. The program may also include a study of the solar heat gain through various heat absorbing glasses, and possibly additional work on condensation.

INSULATION, 15—R. T. Miller, *Chairman;* E. A. Allcut, R. E. Backstrom, Wharton Clay, R. E. Daly, W. A. Danielson, H. C. Dickinson, J. D. Edwards, W. V. Hukill, E. C. Lloyd, Paul McDermott, E. W. McMullen, W. T. Miller, E. R. Queer, T. S. Rogers, F. B. Rowley, W. S. Steele, J. H. Waggoner, G. B. Wilkes.

During the year this committee carefully reviewed that portion of THE GUIDE devoted to heat transmission and made pertinent recommendations to the Guide Publication Committee for suggested revision. In view of the development of a new code for testing insulating materials the committee is making recommendations to the Society that a plan be formulated for retesting all insulating materials of current manufacture.

Considerable thought was also given to the causes of condensation in walls and attics of building structures and a chart has been developed showing permissible inside relative humidities to obviate such difficulties based on outside temperatures and type of wall construction.

SOUND CONTROL, 16—J. S. Parkinson, *Chairman;* C. M. Ashley, A. M. Greene, Jr., A. L. Kimball, V. O. Knudsen, R. F. Norris, C. H. Randolph, J. P. Reis, W. P. Roop, A. E. Stacey, Jr., G. T. Stanton, F. R. Watson.

The primary activity of this committee for the past year has been in connection with the preparation of a standard method for rating apparatus noise. Based on the comments of committee personnel this standard method has been revised and substantial agreement has been obtained on most of the provisions. At the present time further activity is hindered by the fact that some development work is necessary on measuring techniques. A method has been outlined for rating the noise from apparatus in open acoustically treated rooms and for measuring noise within a duct system. However, before final recommendations can be made on certain test procedures further research work is necessary.

A meeting of the committee was held in New York on May 17 and recommendations were made for conducting an investigation on the nature of sound transmission in duct systems without the use of absorbent lining but no suitable location could be determined for conducting the tests. A tentative program was also outlined by the committee for studying the relation of various physical factors in sound to their annoyance but this work has also been held in abeyance due to lacking laboratory facilities and funds to support the project.

COOLING TOWERS, EVAPORATIVE CONDENSERS AND SPRAY PONDS, 17—B. M. Woods, *Chairman;* J. C. Albright, S. C. Coey, E. H. Hyde, E. H. Kendall, S. R. Lewis, J. Lichtenstein, J. F. Park, E. H. Taze.

Work has been continued at the University of California this year through a cooperative agreement with the Society to study the fundamental properties of cooling tower design. A paper on the performance of cooling towers was presented at the 1939 Semi-Annual Meeting and a report is now being completed covering an analytical study of drop dynamics which will be available for the Annual Meeting.

During the past year, preliminary computations have been made on the absorption of solar radiation by means of water films of various thicknesses. Tests have been made giving results of air-liquid surface heat exchangers in which the air side is wetted by means of water. Also observations have been made on free convection from an atmospheric tower and data have been obtained giving the pressure drop of air-foil packed towers, the latter of which will be correlated with heat transfer and mass transfer results. Plans for determination of drop size distribution from jets, characteristics of drift particles, rates of growth and decay of water particles, are progressing. The Committee is also interested in enlarging all of these studies to include some problems of evaporative condensers and roof cooling.

PSYCHROMETRY, 18—F. R. Bichowsky, *Chairman;* D. B. Brooks, W. H. Carrier, H. C. Dickinson, A. W. Gauger, J. A. Goff, William Goodman, A. M. Greene, Jr., L. P. Harrison, F. G. Keyes, D. M. Little, D. W. Nelson, W. M. Sawdon, F. O. Urban.

The major activity of the Committee is an exact determination of the thermodynamic properties of air-water vapor mixtures. Since the laws of combination of air and water vapor deviate from the perfect gas laws by an amount which is of the order of one per cent, this work, of necessity, dictates accurate experimental investiga-

tions. A theory has been developed which is based upon statistical mechanics on the theoretical side and on available data on the equations of state of air and steam on the practical side. The next step is the experimental testing of the theory which is being done under a cooperative agreement with the University of Pennsylvania. The final apparatus is now being constructed and assembled for conducting the investigations. At the request of the Guide Publication Committee, the Committee supervised the redrawing of the Bulkeley Psychrometric Chart to make it consistent with Table 6 in Chapter 1 of THE GUIDE and to improve its accuracy and make it simpler to read.

CORROSION, 19—A. R. Mumford, *Chairman;* H. E. Adams, J. F. Barkley, W. H. Driscoll, T. J. Finnegan, W. Z. Friend, R. R. Seeber, F. N. Speller, A. E. Stacey, Jr.

Activities of a former research committee dealing with corrosion in air conditioning systems were transferred to this committee during the 1939 Annual Meeting. The major research project planned for investigation this year dealt with the determination of the amount of non-condensable gases that will dissolve in the condensate formed in a heating system at different rates of condensation. This work was being studied through a cooperative agreement at the Michigan College of Mining and Technology but was interrupted during the summer by a disastrous fire. The Committee authorized the continuation of the project and the contributors agreed so steps are now being taken at the College to re-equip a new laboratory.

A joint session of the members of the *National Warm Air Heating and Air Conditioning Association* and the ASHVE was held at the Hollenden Hotel at 2:00 p.m. on Wednesday, January 24. Pres. L. R. Taylor, *NWA&ACA*, called the meeting to order and presented the gavel to J. F. McIntire, president of the ASHVE. He then introduced W. L. Fleisher, chairman of the ASHVE Committee on Research, who gave a brief outline of the program being carried on by the Society. He pointed out some of the practical applications of research results which have been developed since the establishment of the Society's Research Laboratory 20 years ago. The National Warm Air Research sponsored by the Association was outlined by F. L. Meyer in a brief talk.

An interesting address was delivered by C. H. Mylander, banker, Columbus, Ohio.

The last technical session convened at 10:00 a.m. on Thursday, January 25, with President McIntire presiding. He introduced T. M. Dugan, McKeesport, Pa., who represented the ASHVE on Committee B-16 of the *American Standards Association,* and who submitted a written report which he outlined briefly at this session.

There was an intermission from the technical session and President McIntire discussed the progress of the membership in the Society during the past year, and brought out the fact that he believed a steady growth was much better for the Society than membership drives that would bring in members who would stay for a year and drop out. He mentioned those members who were present at the meeting who joined the Society 25 years ago, and called upon W. H. Driscoll, past president of the Society, to escort the incoming president to the chair.

New Officers Installed

The newly elected officers who were inducted into office were: Pres. F. E. Giesecke, College Station, Tex., 1st Vice-Pres. W. L. Fleisher, New York, 2nd Vice-Pres. E. O. Eastwood, Seattle, Treas. M. F. Blankin, Philadelphia,

Pa. The four new members of the Council were introduced as follows:
J. F. Collins, Jr., Pittsburgh, E. N. McDonnell, Chicago, T. H. Urdahl,
Washington, D. C., and C.-E. A. Winslow, New Haven.

H. H. Erickson, Philadelphia, chairman of the Committee to Study Method
of Selecting Society Officers and Council Members then presented his report
on procedures for the Nominating Committee, which he moved be sent to the
membership for consideration before final adoption. The motion was seconded
and unanimously adopted.

The report of the Resolutions Committee was then presented by A. F. Nass,
Pittsburgh, and the following resolutions were offered for adoption:

Resolutions

Whereas, the 46th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND
VENTILATING ENGINEERS held in the city of Cleveland has been an outstanding success,
be it hereby resolved that the following resolutions of thanks and appreciation be
spread on the minutes of the Society and copies thereof be transmitted to each of
those persons and agencies contributing thereto.

The City of Cleveland for its hospitality.

The Honorable Harold L. Burton, Mayor of the City of Cleveland, for his cordial
greeting.

The Cleveland Convention and Visitors Bureau for its cooperation and assistance.

The Committee on Arrangements of the Cleveland Chapter for its finely planned
program.

The City press and trade papers for their generous coverage of our sessions.

The Management of the Exposition for the largest show of heating, ventilating and
air conditioning equipment.

The authors of technical papers for their fine contributions.

The Speakers who have addressed us so entertainingly.

To the Hotel Management and staff for their excellent service, and

To all those who have contributed so much to our pleasure and enjoyment while
we have been in Cleveland.

It was moved and seconded, that the resolutions be unanimously adopted.

President Giesecke called for any new business, and there being no further
business, declared the 46th Annual Meeting adjourned at 11:40 a.m.

Exposition and Entertainment

The entire first day, January 22, was devoted to committee meetings, Con-
ference of Chapter Delegates, and a meeting of the Council.

Promptly at 9:30 a.m. the Conference of Chapter Delegates convened in
the Pine room and representatives of all 29 Chapters were present when
President McIntire gave a brief speech of welcome. Earle W. Gray, Okla-
homa City, served as chairman and S. W. Boyd, Atlanta, was secretary of
the Conference.

At 2:00 p.m. the Officers and Council assembled at Lakeside Hall for the
formal opening of the 6th International Heating and Ventilating Exposition,
at which 303 exhibitors participated, and which during the week attracted
the attention of over 25,000 people.

On Wednesday evening the 46th annual banquet of the Society was held in
the grand ballroom of the Statler. At the conclusion of an excellent dinner,
toastmaster Walter Klie introduced W. H. Driscoll, who presented the retiring

president, J. F. McIntire, with the past president's emblem. Mr. McIntire responded briefly and the president-elect, F. E. Giesecke, was then introduced. Two brief, but entertaining talks were given by Dr. W. E. Wickenden, president, Case School of Applied Science, and Dr. Harvey N. Davis, president, Stevens Institute of Technology, Hoboken, N. J., whose address covered the subject of The Engineer's Job Today.

On Thursday many members and guests attended the luncheon of the Rotary Club of Cleveland, and heard Dr. C.-E. A. Winslow, vice-chairman of the Committee on Research, talk on The Air-Cooled Human Body. His address, which was broadcast, was heard by 300 Rotarians and their guests.

PROGRAM 46TH ANNUAL MEETING

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
HOTEL STATLER, CLEVELAND, OHIO
JANUARY 22-26, 1940

COMMITTEE MEETINGS

Sunday, January 21

2:00 P.M. Committee on Treatment of Disease (Parlor B)

Monday, January 22

8:30 A.M. Committee on Air Distribution and Air Friction (Parlor M)
9:30 A.M. Conference of Chapter Delegates (Pine Room)
10:00 A.M. Council Meeting (Tavern Room)
10:00 A.M. Committee on Radiation and Comfort (Parlor F)
10:00 A.M. Committee on Corrosion (Parlor G)
10:00 A.M. Committee on Insulation (Parlor H)
2:00 P.M. Committee on Air Conditioning Requirements of Glass (Parlor H)
2:00 P.M. Committee on Heat Transfer of Finned Tubes (Parlor G)
2:00 P.M. Committee on Heat Requirements of Buildings (Parlor F)
2:00 P.M. Committee on Psychrometry (Tavern Room)
2:30 P.M. Nominating Committee Meeting (Pine Room)
4:00 P.M. Committee on Summer Air Conditioning for Residences (Parlor D)
4:00 P.M. Committee on Weather Design Conditions (Parlor L)
8:00 P.M. Final Meeting 1939 Committee on Research (Tavern Room)

Tuesday, January 23

12:00 NOON Committee on Solid Fuels (Parlor B)
8:00 P.M. Organization Meeting 1940 Committee on Research (Tavern Room)

Thursday, January 25

1:30 P.M. Council Meeting—Hotel Statler (Tavern Room)

SPECIAL EVENTS FOR LADIES

Monday, January 22

8:30 A.M. Ladies Reception (Parlors 1 and 2—Mezzanine)

Tuesday, January 23

9:30 A.M. Ladies Tour—Cleveland Orchestra Rehearsal—Severance Hall
12:00 NOON Ladies Luncheon—Crosby's Restaurant (East 105th and Carnegie)

1:30 P.M.   Ladies Tour—Electrical Housekeeping Institute, Nela Park (Trip starts from Crosby's)

8:00 P.M.   Informal Theater Parties

Wednesday, January 24

10:00 A.M.   Ladies—Open for Shopping

2:30 P.M.   Ladies Bridge—Halles Tea Room

4:00 P.M.   Ladies Tea and Fashion Show—Halles Tea Room

TECHNICAL SESSIONS

Monday, January 22

8:30 A.M.   Registration—Hotel Statler (Mezzanine)

9:30 A.M.   Conference of Chapter Delegates (Pine Room)

2:00 P.M.   Opening of 6th International Exposition—Lakeside Hall

5:00 P.M.   Social Hour (Pine Room—Mezzanine)

10:00 P.M.   Get-together Party (Euclid Ballroom)

Tuesday, January 23

8:30 A.M.   Registration—Hotel Statler (Mezzanine)

9:30 A.M.   Hotel Statler (Grand Ballroom)
Greetings by Hon. Harold L. Burton, Mayor of Cleveland
Response by Pres. J. F. McIntire
Reports of Officers
Reports of Council Committees
Technical Papers:
Effect of Lint on Air Filter Performance, by Frank B. Rowley, and Richard C. Jordan
Gas Equipment for Year-Round Air Conditioning, by G. E. May
Dynamic and Thermal Behavior of Water Drops in Evaporative Cooling Processes, by H. B. Nottage and L. M. K. Boelter
Report of Tellers of Election

12:00 NOON   Heating and Ventilating Exposition—Lakeside Hall

2:00 P.M.   Hotel Statler (Grand Ballroom)
Heat Gain Through Glass Blocks by Solar Radiation and Transmittance, by F. C. Houghten, David Shore, Harold T. Olson and Burt Gunst
Thermal Test Coefficients of Aluminum Insulation for Buildings, by G. B. Wilkes, F. G. Hechler and E. R. Queer
Diminishing Effectiveness of Successive Thicknesses of Insulating Materials, by Paul D. Close
Calculation of Coil Surface Areas for Air Cooling and Dehumidification, by John McElgin and D. C. Wiley

5:00 P.M.   Social Hour (Pine Room—Mezzanine)

6:30 P.M.   Past Presidents' Dinner

8:00 P.M.   Informal Theater Parties

10:00 P.M.   Stag Smoker—Hollenden Hotel (Grand Ballroom)—Sponsored by the National Warm Air Heating and Air Conditioning Association for their members and guests and those of the ASHVE. A colorful event after Exposition hours—Speechless—Informal—Secure tickets at Registration Desk

Wednesday, January 24

9:30 A.M.   Hotel Statler (Grand Ballroom)
Report of Committee on Research, W. L. Fleisher, Chairman
The Peripheral Type of Circulatory Failure in Experimental Heat Exhaustion—The Role of Posture, by R. W. Keeton, F. K. Hick, Nathaniel Glickman and M. M. Montgomery

Advantages of Bactericidal Ultraviolet Radiation in Air Conditioning Systems, by H. C. Rentschler and Rudolph Nagy

Seasonal Variation in Reactions to Hot Atmospheres, by F. C. Houghten, A. A. Rosenberg and M. B. Ferderber

12:00 NOON Heating and Ventilating Exposition—Lakeside Hall

2:00 P.M. Hotel Hollenden (Grand Ballroom)—Joint Meeting with National Warm Air Heating and Air Conditioning Association—Co-chairmen, Pres. J. F. McIntire, ASHVE, and Pres. L. R. Taylor, NWAHACA

Address by Charles H. Mylander, Columbus, Ohio

National Warm Air Research, by F. L. Meyer

ASHVE Research, by W. L. Fleisher

Performance of a Stoker-Fired Warm-Air Furnace as Affected by Burning Rate and Feed Rate, by A. P. Kratz and S. Konzo

7:00 P.M. 46th Annual Banquet—Hotel Statler (Grand Ballroom)—Presentation of Past President's Emblem—Toastmaster—Walter Klie; Speakers —Dr. W. E. Wickenden, President, Case School of Applied Science, Dr. Harvey N. Davis, President, Stevens Institute of Technology— Address: The Engineer's Job Today

Thursday, January 25

10:00 A.M. Hotel Statler (Euclid Ballroom)

Analysis of The Factors Affecting Duct Friction, by J. B. Schmieler, F. C. Houghten and Harold T. Olson

The Performance of Stack Heads, by D. W. Nelson, D. H. Krans and A. F. Tuthill

Installation of Officers

12:00 NOON Heating and Ventilating Exposition—Lakeside Hall

12:00 NOON Rotary Club Luncheon (Grand Ballroom Hotel Statler) for Rotarians and Guests—Speaker: Dr. C.-E. A. Winslow. Subject: The Air Cooled Human Body

1:00 P.M. Industrial Rayon Corp.†—Painesville † (Windowless completely conditioned plant) Transportation $1.00 per person (round trip)

1:30 P.M. Inspection Trips †

American Gas Association Laboratory

Case School of Applied Science Laboratory

Republic Steel Corp.

Friday, January 26

12:00 NOON Heating and Ventilating Exposition—Lakeside Hall

COMMITTEE ON ARRANGEMENTS

L. T. AVERY, *General Chairman;* G. L. TUVE, *Honorary Chairman;* PHILIP COHEN, *Vice-Chairman;* L. S. RIES, *Vice-Chairman.*

Technical Sessions: C. F. Eveleth, *Chairman;* W. E. Stark.

Publicity: John Paul Jones, *Chairman;* C. M. H. Kaercher, W. R. Moore.

Finance: E. W. Gray, *Chairman;* Walter Klie, R. A. Wilson.

Inspection Trips: C. A. McKeeman, *Chairman;* W. R. Beach, G. P. Nachman, E. F. Steffner, N. H. Hall.

Attendance: D. L. Taze, *Chairman;* J. L. Berger, H. F. Curtis, D. E. Mannen, Jr.

Banquet: F. A. Kitchen, *Chairman;* Mrs. F. A. Kitchen,* G. G. Auer, R. G. Davis, W. A. Evans, W. M. Rowe, D. K. Wright, Jr.

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† For Ladies' Trip Tuesday and Men's Inspection Trips sign up for transportation at Registration Desk. Limit of 30 persons for the inspection trip to Industrial Rayon due to necessity of guides.

* Member of Ladies Committee.

Transportation: A. L. Vanderhoof, *Chairman;* W. R. Beach, M. I. Levy, W. R. Rhoton, J. A. Schurman, L. O. Weldy, J. M. Black.

Entertainment: H. E. Wetzel, *Chairman;* Mrs. H. E. Wetzel,* Walter Baggaley, E. B. Cary, R. L. Clark, E. J. Sable, R. T. Southmayd.

Registration-Reception: P. D. Gayman, *Chairman;* C. F. Cushing, J. H. Ferguson, S. R. Gilbert, H. W. Heisterkamp, D. E. Humphrey, W. F. Jackson, J. V. Koubeck, G. B. Longcoy, J. L. Mauer, H. M. Nobis, W. L. Norrington, H. L. Repp; L. E. Slawson, Kelvin Trenner, J. E. Wilhelm, E. O. Young; and Mesdames P. D. Gayman,* Walter Baggaley,* E. B. Cary,* W. A. Evans,* D. E. Humphrey,* John Paul Jones,* C. M. H. Kaercher,* G. P. Nachman,* J. A. Schurman,* L. O. Weldy,* D. K. Wright, Jr.,* C. A. McKeeman.*

Ladies: Mrs. W. R. Rhoton, *Chairman;* Mrs. G. L. Tuve, *Vice-Chairman;* Mrs. L. T. Avery, *Vice-Chairman;* Mrs. W. R. Beach, Mrs. C. F. Eveleth, Mrs. E. W. Gray, Mrs. D. L. Taze, Mrs. A. L. Vanderhoof.

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* Member of Ladies Committee.

THE EFFECT OF LINT ON AIR FILTER PERFORMANCE

By FRANK B. ROWLEY * AND RICHARD C. JORDAN,** MINNEAPOLIS, MINN.

I N making laboratory tests on air filters it has been the common practice to use a dust mixture which has been made up from several distinct types of dusts. The elements have usually included some type of carbon black or lampblack in combination with the ash from a specified coal. Other elements such as volcanic ash, pulverized clay, and Fuller's earth have also been used in the dust mixture, and in many cases a single element such as coal ash or some other dust has been used individually. The properties of these materials which affect the performance of an air filter vary through wide ranges, and many laboratory tests have shown that not only are the performance characteristics of a given filter affected by the type of dust used, but that the relative values of two different filters may even be reversed when tested by the same laboratory apparatus and test procedures but with different dust mixtures. The variables due to the different types of dust have been investigated and the results reported in three previous papers.[1, 2, 3]

In practically all previous investigations the dust mixture has contained no lint or fibrous matter, whereas in practice nearly all air to be filtered contains some form of fibrous material, commonly called lint. Fibrous materials will often have a greater effect on the performance characteristics of a filter than will the straight dust mixtures commonly used. An examination of the used filters from many installations will show a mat of lint at the entrance of the filter with very little dust distribution through the filter.

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* Director, Engineering Experiment Station, University of Minnesota. MEMBER ASHVE.
** Instructor, Engineering Experiment Station, University of Minnesota. MEMBER ASHVE.
[1] ASHVE RESEARCH REPORT No. 1094. Air Filter Performance as Affected by Kind of Dust, Rate of Dust Feed and Air Velocity Through Filter, by Frank B. Rowley and Richard C. Jordan. (ASHVE TRANSACTIONS, Vol. 44, 1938, p. 415.)
[2] ASHVE RESEARCH REPORT No. 1122. Air Filter Performance as Affected by Low Rate of Dust Feed, Various Types of Carbon, and Dust Particle Size and Density, by Frank B. Rowley and Richard C. Jordan. (ASHVE TRANSACTIONS, Vol. 45, 1939, p. 339.)
[3] ASHVE RESEARCH REPORT No. 1143. A Standard Air Filter Test Dust, by Frank B. Rowley and Richard C. Jordan. (ASHVE TRANSACTIONS, Vol. 45, 1939, p. 681.)
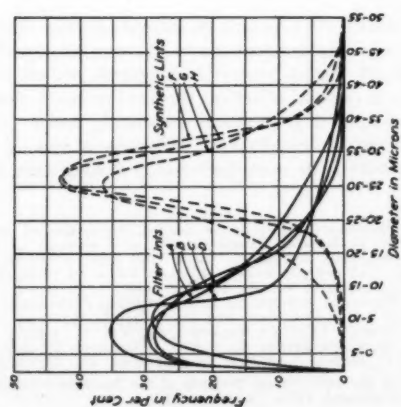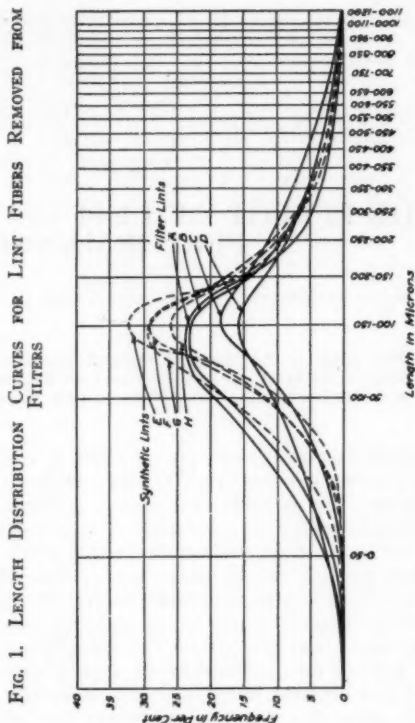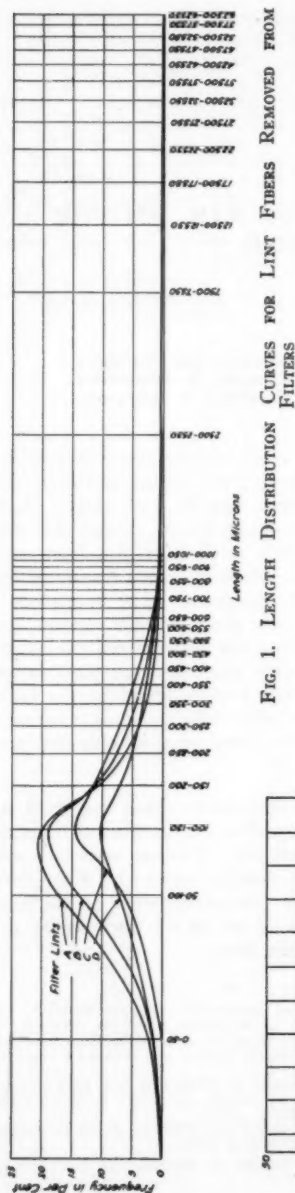
FIG. 1.   LENGTH DISTRIBUTION CURVES FOR LINT FIBERS REMOVED FROM FILTERS

FIG. 2.   DIAMETER DISTRIBUTION CURVES FOR FILTER LINT FIBERS AND SYNTHETIC LINT FIBERS

FIG. 3.   LENGTH DISTRIBUTION COMPARISON FOR SYNTHETIC LINT FIBERS AND FILTER LINT FIBERS SHORTER THAN 1200 MICRONS

TABLE 1.   DISTRIBUTION OF FIBER SIZE OF LINTS REMOVED FROM FILTERS IN ACTUAL USE

SAMPLE	SOURCE OF LINT	PER CENT OF FIBERS GREATER THAN 1200 μ LENGTH	PER CENT OF WEIGHT DUE TO FIBERS GREATER THAN 1200 μ LENGTH	AVERAGE DIAMETER IN MICRONS	AVERAGE LENGTH IN MICRONS	AVERAGE LENGTH IN MICRONS OF ALL FIBERS SHORTER THAN 1200 μ	AVERAGE LENGTH IN MICRONS OF ALL FIBERS LONGER THAN 1200 μ
A	Office Bldg. Filter	17.0	77.0	10.8	996	271	4,420
B	Office Bldg. Filter	18.4	86.0	10.5	1,415	251	6,500
C	Suburban Residential Filter	22.4	90.0	10.7	2,702	378	10,920
D	Pullman Car Filter	35.0	92.0	13.2	2,923	364	7,560

There are many variables in the lint problem such as type of lint, length and diameter of fibers, density, percentage of lint by weight in the total dust in the air, etc. It is, therefore, not possible to get a positive answer which can be applied to all installations, but lint is, nevertheless, a vital factor in the performance characteristics of filters in many practical installations and should be taken into consideration in rating them.

In this investigation there were three distinct problems to consider: *first,* to determine the character of the lint which must be taken out of the air by the average filter installation; *second,* to produce a lint which will simulate as nearly as possible that found in the air and which can be reproduced with reasonable certainty without undue expense; *third,* to study the effect of the lint produced on the performance characteristics of different types of filters.

TYPE OF LINT IN AIR

An inspection of certain types of filters which have been subjected to air containing lint will show a heavy mat of lint on the entering face of the filter with very little penetration to the interior parts of the filter. It is evident that the lint in this mat, particularly that which is collected on the entering surface near the end of the life of the filter, is representative of that found in the air. Use was made of this principle in selecting samples of atmospheric lint for examination. Four filters, each of which had been in use for a sufficient length of time to be covered on the entering surface with a heavy mat of lint, were selected. Lint samples were taken from them as follows:

A—Lint taken from filter in downtown office building.
B—Same as A but taken from different air conditioning system located on different floor of same building.
C—Lint taken from residential filter, suburban district.
D—Lint taken from Pullman car filter.
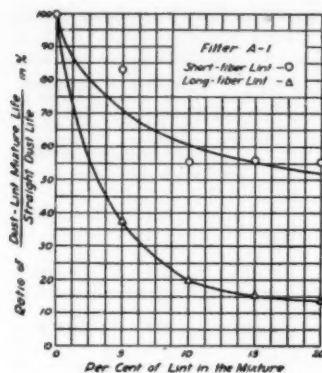
FIG. 4. EFFECT OF DUST-LINT RATIO
ON LIFE OF FILTER *A-1*
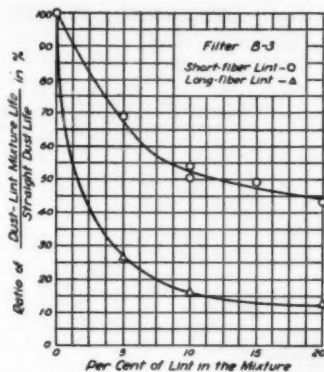
FIG. 5. EFFECT OF DUST-LINT RATIO
ON LIFE OF FILTER *B-3*

In each case the sample was removed from the entering surface of the mat, and the mat was of sufficient thickness to insure that no lint could penetrate the filter to any appreciable depth.

In preparing a sample for analysis a small portion of the lint was held over a glass slide and pulled apart to separate completely all fibers. A cover glass was then placed over the fibers, and the analysis was made with a microscope of 100 diameters magnification and a ruled ocular in the eyepiece. The diameters and lengths of from 200 to 400 fibers of each type of lint were measured as closely as possible by the ruled ocular and recorded.
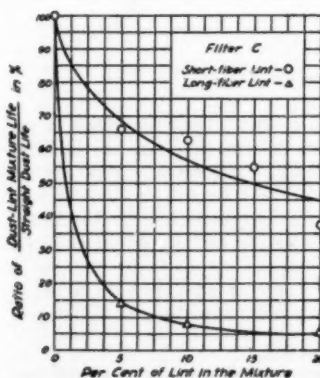
FIG. 6. EFFECT OF DUST-LINT RATIO
ON LIFE OF FILTER *C*
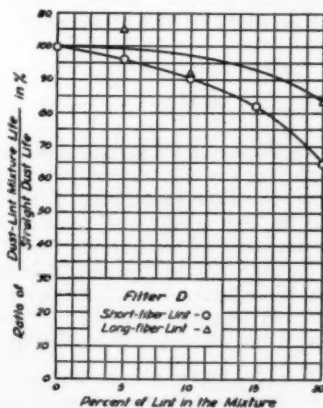
FIG. 7. EFFECT OF DUST-LINT RATIO
ON LIFE OF FILTER *D*

These data were then grouped into length intervals of 50 μ (microns) and diameter intervals of 5 μ, and the frequencies with which the fibers occurred in each interval calculated. Naturally, there was some dust in the lint samples, but it was comparatively easy to distinguish between the dust particles and the fibers.

The results of the microscopic analysis of the fibers taken from the various filters are shown in the curves of Figs. 1 and 2 and in Table 1. Frequency

TABLE 2.   EFFECT OF DUST-LINT RATIO ON FILTER LIFE AND ARRESTANCE

TYPE OF LINT	PER CENT by WEIGHT OF LINT IN Mixture	FILTER A-1		FILTER B-3		FILTER C		FILTER D	
		Arrestance in Per Cent	Ratio[a] $\frac{A}{B}$ in Per Cent	Arrestance in Per Cent	Ratio[a] $\frac{A}{B}$ in Per Cent	Arrestance in Per Cent	Ratio[a] $\frac{A}{B}$ in Per Cent	Arrestance in Per Cent	Ratio[a] $\frac{A}{B}$ in Per Cent
	0	81.0	100.0	87.2	100.0	70.5	100.0	92.4	100.0
	5	83.7	83.4	89.5	68.9	76.5	66.1	93.0	96.1
Short Fiber Lint	10 ...	84.7 ...	55.6 ...	88.7 89.6	50.2 53.7	76.8 ...	62.4 ...	90.6 93.5	91.0 90.4
	15	85.8	56.0	89.5	49.3	76.6	54.8	93.7	82.0
	20	86.3	55.4	89.4	43.2	78.3	37.4	93.4	64.8
	0	81.0	100.0	87.2	100.0	70.5	100.0	92.4	100.0
	5	85.7	37.9	91.7	26.6	71.6	13.9	93.0	105.0
Long Fiber Lint	10	88.7	19.7	92.0	16.0	70.6	7.8	93.4	91.7
	15	86.4	15.2	...	...	...	...	...	...
	20	88.8	13.6	93.2	12.4	76.5	5.2	95.5	83.3
14 per cent short fiber 86 per cent long fiber	10	86.5	21.2	90.7	16.0	...	...	...	...

[a] A = Filter life in hours when tested on dust-lint mixture.
B = Filter life in hours when tested on dust mixture alone.

plotted against length in microns is shown in the curves of Fig. 1. Due to the facts that the scale for length in microns was extended and that the greater frequencies were in the low range, the length in microns was plotted to a logarithmic scale. It will be noted from these curves that fibers from 100 to 150 μ in length occur in the greatest frequency for all samples. Above 1200 μ there is substantially a linear distribution. The curves of Fig. 1 do not indicate the maximum fiber length for each sample examined since the curves practically join above 1200 μ in length. The longest fibers recorded were 24,317, 29,713, 57,036, and 51,030 μ for samples A, B, C, and D, respectively.

The frequency in per cent plotted against diameter in microns for the four samples is shown in the curves of Fig. 2. These curves show that for all samples the maximum diameter frequency is from 5 to 10 μ. Table 1 gives the source and the analysis of the four lint samples. From the curves and the table the following observations can be made: *First*, for all samples the greatest percentage of fibers by length was less than 1200 μ, but the greatest percentage of weight for the entire sample was due to fibers of more than 1200 μ in length. The average measurements for all samples
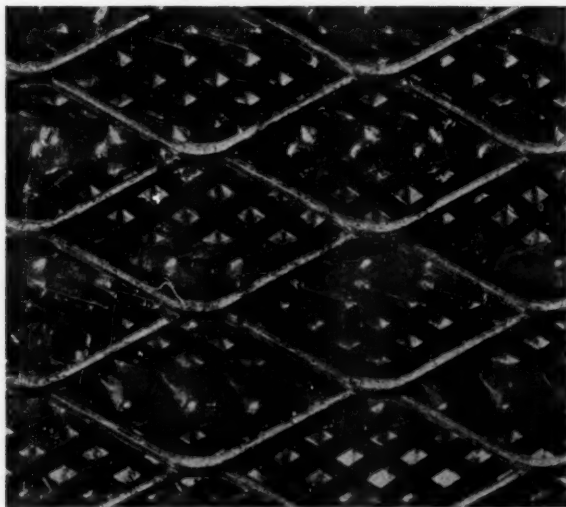
FIG. 8.   FILTER *A-1* BEFORE AND

showed that 23 per cent of all fibers was more than 1200 μ in length and that these fibers constituted 86 per cent of the entire weight of the samples. *Second,* the variation in average lint diameter for the four samples was from 10.8 to 13.2 μ. *Third,* the percentage of fibers of greater length than 1200 μ was lowest for the office building, next for the residence, and highest for the Pullman car. The average diameter of fibers for the Pullman was likewise the highest, but the other three were substantially the same. The average length in microns for all fibers was lowest for the office buildings, next for the residence, and highest for the Pullman. However, there was very little difference between the lint samples from the residence filter and those from the Pullman filter. These differences in size between the fibers found in filters for the three different types of installations may be due partly to the average distances between the source of lint and the filters, but are probably largely

due to the type of lint generated in the three different installations. There is, however, not a great difference considering the widely different character of the installations. *Fourth,* from the curves it will be noted that there is a much wider spread in fiber length than in fiber diameter for all samples. It should be noted that from 200 to 400 fibers do not give a sufficient sampling to obtain statistically accurate distribution curves and averages. However, the results are sufficiently accurate for purposes of comparison between actual lint found in the air and the artificial lint to be built up for test purposes.
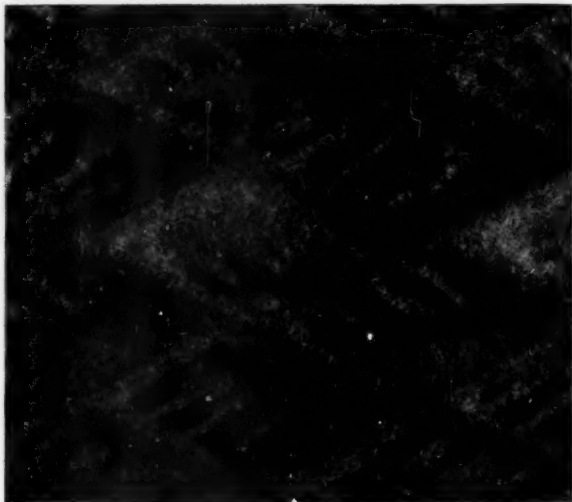
AFTER LABORATORY LINT TEST

ARTIFICIAL LINT

In producing an artificial lint to simulate that found in the atmosphere it was necessary to consider both length and diameter of lint fibers. Several different types of materials were considered, including wool, cotton, silk, rayon, and kapok. In general the fibers of these materials were too long and of too large a diameter when compared with those taken from filters. After a considerable amount of experimental work it was found practical to duplicate approximately the long fibers, that is those over 1200 μ in length, by combing out a mat of the fibers and cutting it into $\frac{3}{16}$-in. lengths with a paper cutter. For the short fibers it was found that kapok, when ground in a ball mill for an experimentally determined period of time, broke up into lengths which conformed very closely in frequency distribution of length

with the short fibers taken from filter installations. The technique consisted of grinding the fibers for a period of 20 min in a ball mill rotating at 60 rpm. The barrel of the mill was $8\frac{1}{4}$ in. in diameter, and $8\frac{3}{4}$ in. in length, and contained 150 flint stones of 1-in. average diameter. Four grams of Java kapok were ground and then screened through a 30-mesh screen to remove the coarse, unbroken fibers.

Four samples of Java kapok, designated as E, F, G, and H, were obtained from different sources and all treated in the manner as previously outlined.

FIG. 9.   FILTER B-3 BEFORE AND

After treatment these samples were examined microscopically for diameter and length distribution. The lengths were found to be all under 1200 μ and the greatest percentage of fibers was from 100 to 150 μ in length. The length frequency distribution curves for the kapok fibers and for the filter lint fibers below 1200 μ are shown in Fig. 3. From these curves it will be noted that there is a reasonably close agreement in length frequency distribution between the artificially prepared kapok fibers and the filter fibers below 1200 μ in length. The diameter distribution of the prepared kapok fibers is shown in Fig. 2 in conjunction with the diameter distribution for those fibers taken from filters. It will be noted that the average diameter of kapok fibers is considerably larger than that found in the lint from the filters. There is, however, a very close agreement in the diameter distribution for the fibers from the different samples of kapok lint prepared.

In the ball mill the kapok fibers were broken into shorter lengths, but the

diameters were not materially changed. The diameter distribution for the short kapok fibers as shown in Fig. 2 may, therefore, be used as the diameter distribution for the long kapok fibers which were prepared by cutting pads of the kapok as previously described. It was found impractical to prepare any of the other types of lint in the ball mill due to the fact that they were not broken up by the grinding process to an extent equal to that for the kapok fibers. The kapok fibers were found to be rather brittle and easily broken. A criticism of the prepared kapok lint as a test fiber is in the fact that it is
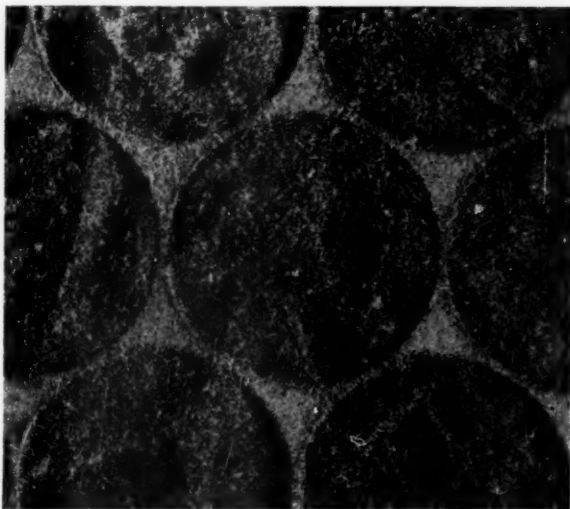
AFTER LABORATORY LINT TEST

of greater diameter than that found in the average filter. The effect of lint on the filter, however, appears to be more a question of fiber length than of fiber diameter, providing the fiber is of a size and density which will float in the air, and will be distributed evenly over the face of the filter. For this reason it seems practical to use the prepared kapok fibers in determining the relative performance of filters with and without lint in the air.

FILTER TEST DATA

As in the previous program, four filters typical of those used in practice were used throughout all of the tests. These filters have been designated by the letters *A-1, B-3, C* and *D,* and their construction was described in detail in a previous paper.[4] A numeral following the designating letter indicates

4 Loc. Cit. Note 1.

that slight changes had been made by the manufacturers in the filter since it was originally selected. Briefly, the filters may be described as follows:

A-1—A cleanable type oiled filter, 2 in. thick. The filter media was made up of layers of expanded metal and wire screen graded from coarse mesh at the entrance to fine mesh at the leaving side. The filter is 2 in. in thickness, whereas the former filters of this type used in the previous test program were 4 in. in thickness.

B-3—A viscous throw-away type of filter, 2 in. thick. The fibrous media were graded in fiber size, density, and oiling from the entering to the leaving side. Slight
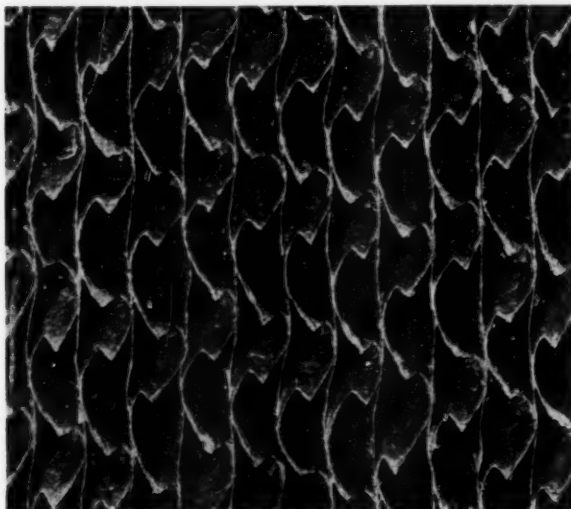
FIG. 10.   FILTER *C* BEFORE AND

changes were made by the manufacturer in this filter since the original selection was made.

C—A cellular type of filter, 2 in. in thickness. The filter was built of two sections, each with honeycomb type of cells set at 45 deg to the center line of the duct, and at 90 deg to each other. The cells in the section on the entering side were of larger dimension than those in the section on the leaving side of the filter.

D—A filter of cotton media consisting of coarse material on the entering side and glazed on the leaving side. The filter media was accordion plaited in the frame to give an area of approximately 12 times the cross-sectional area of the air stream.

In testing these filters both long and short fiber lints were used, and these were mixed with various proportions of dust as indicated in the test results.

In all cases the dust mixture consisted of 80 per cent by weight of Pocahontas ash screened through a 200-mesh screen, and 20 per cent by weight of

Double Bolted Carbon Dust screened through 100 mesh.   The Pocahontas
ash was obtained by burning Pocahontas coal at a rate which was sufficiently
low to prevent clinkering of the ash, and the Double Bolted Carbon Dust was
obtained from Binney and Smith Co., New York City.   When the short fiber
lint was used it was mixed with the test dust and fed to the filter by the
standard dust feeding apparatus.   When the long fiber lint was used, it was
necessary to feed it into the air stream separately from the dust.   This was
accomplished by placing the lint in a copper mesh cylindrical shell closed at
both ends and rotated in the inlet air stream leading to the filter.   As the
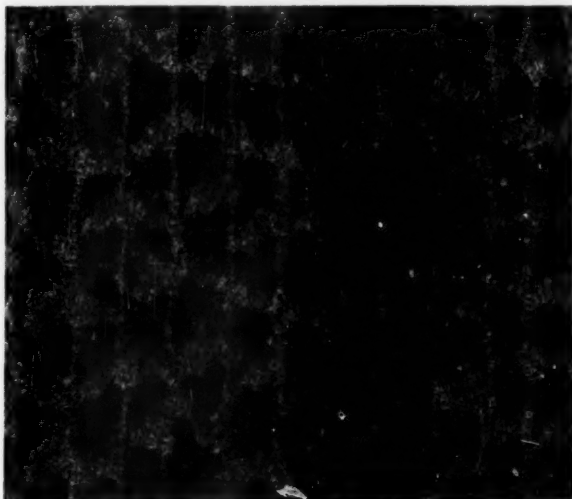
AFTER LABORATORY LINT TEST

cylinder containing the lint was rotated, a jet of compressed air was directed
tangentially against the cylindrical copper mesh shell, thus forcing the fibers
through the mesh of the screen.   The amount of lint for a 1-hour test was
divided into two parts, ½ being fed into the air stream at the end of the
first 15 min of the test, and the second half at the end of the first 45 min
of test.   Approximately 2 or 3 min were required to feed the lint at each
interval.   In each test the face air velocity through the filter was maintained
at 300 fpm, and the total combined weight of the dust and lint fed was 40 g
(grams) per hour.   In all cases the length of the test was governed by the
time required for the filter resistance to rise 0.25 in. of water.   Preliminary
tests were made using Java kapok obtained from three different sources in
order to determine the uniformity of this material as a test lint.   The varia-
tions in test results were not appreciably greater than those for a series of

tests run on kapok obtained from a single source and tested on the same make of filter.

Tests were run on all four types of filters, using both short fiber and long fiber kapok at percentages of lint ranging from 0 to 20 per cent of the combined weight of lint and dust. Filters *A-1* and *B-3* were also tested on a mixture of lint consisting of 14 per cent by weight of the short lint and 86 per cent by weight of the long lint. This mixture approximated the weight ratio of the two lengths as found in actual installations, and given in Table 1.
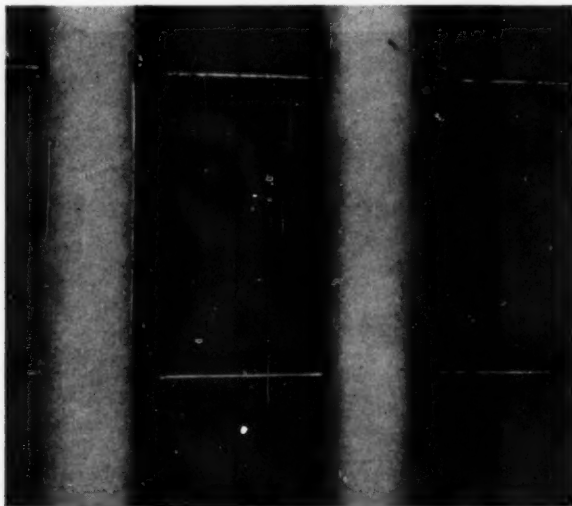
FIG. 11.   FILTER *D* BEFORE AND

The results of the tests are shown in Table 2, and graphically in Figs. 4, 5, 6 and 7 for Filters *A-1*, *B-3*, *C*, and *D*, respectively. The curves show that, with the exception of Filter *D*, there is a rapid drop in life for small percentages of lint mixed with the dust. In each case the long fiber lint affects the life to a much greater extent than the short fiber lint for the same percentage mixture. For Filter *D* the effect of lint on filter life was very much less pronounced than for the other filters. With this filter the actual surface area for catching lint was about 12 times as great as that for the other filters, and since the deposit of lint was mainly on the surface of the filter the results are logical. Furthermore, the lint was of the same general type of material as the filtering media, and lint deposits on the face, therefore, merely had the effect of increasing the thickness of the media. The reason that the short fibers do not affect the life of the Filters *A*, *B*, and *C* to the same extent as the long fibers is that there is better distribution of the short fibers in the interior of the filters, whereas the long fibers tend to mat at a particular

section of the media and cause abnormally high pressure drops across such sections.

From Table 2 it will be noted that there is a tendency for an increased arrestance with increasing percentages of lint in the dust mixture. This tendency is somewhat greater for Filters *A-1* and *C* than for Filters *B-3* and *D*. A part of this increased arrestance is probably due to the increase in dust eliminating efficiency of the media when containing lint, and a part to the fact that most filters are practically 100 per cent efficient in the elimination
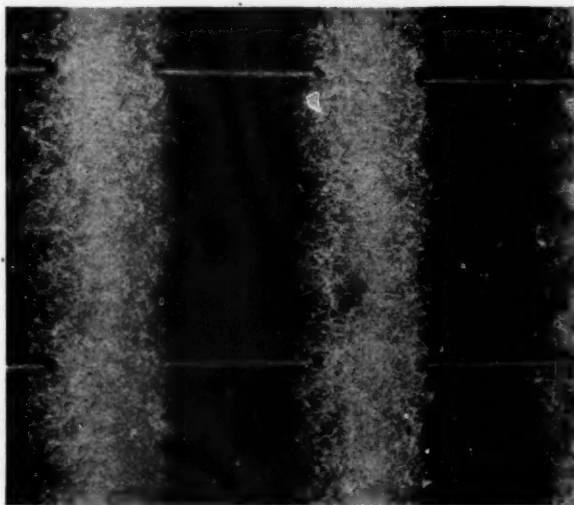
AFTER LABORATORY LINT TEST

of lint. All arrestances shown are based upon the total weight of dust and lint fed. The arrestance values are substantially the same when using the short fiber lint as when using the long fiber lint for a particular filter.

When 14 per cent of short fibers was mixed with 86 per cent of long fibers, and 10 per cent by weight of this fiber mixture combined with 90 per cent by weight of the dust mixture, the performance characteristics of Filters *A-1* and *B-3* as shown in Table 2 were substantially the same as those for the same filters when using a mixture consisting of 10 per cent of long fibers and 90 per cent of dust. These results are logical since there is about six times as much long fiber by weight as short fiber in the mixture. The probable effect of this preponderance of long fibers is that a part of the short fibers are caught on the surface of the filter by the long fibers and are thus prevented from penetrating to the depth to which they might go if short fibers only were used in the mixture. These tests indicate, therefore, that a relatively long fiber lint might be used satisfactorily for the rating of air filters.

TABLE 3.   DISTRIBUTION OF LONG LINT FIBERS IN FILTERS AFTER
LABORATORY TESTS

FILTER	DISTRIBUTION OF LINT IN FILTER	AMOUNT LINT PASSED THROUGH FILTER
A-1	Lint deposited throughout filter with some of the fine mesh openings at exit of filter completely plugged	None
B-3	Lint matted across inlet surface of filter—practically all lint removed in first ½ in. layer of filtering media	None
C	Majority of lint deposited at entrances to first and second passes—heaviest deposit at second pass with some cells entirely plugged	Trace
D	Lint deposited between fingers with space plugged practically solid 4 to 5 in. back from exit side of filter	None

Figs. 8, 9, 10, and 11 show comparisons of the four filters before and after tests with long fiber lint-dust mixture. Table 3 gives a comparison of the distribution of lint in the four filters as determined by examination after the tests. An examination of the filters showed a reasonably even distribution of lint over the face of the 2-in. filters for both the short and long fibers, indicating good distribution of the lint in the test air. In all cases the deposits adjacent to the retaining frames were slightly greater due to the area restrictions at these points. With the exception of Filter C, all lint appeared to be retained in the filter. For Filter C a slight trace of very short fiber lint was found in the crucible on the down stream side of the filter.

CONCLUSIONS

The conclusions which may be drawn from this investigation are as follows:

1. The lint in the air contains both short and long fibers, the greater percentage by number being the short fibers, but the greatest percentage in weight being the long fibers. From the tests made it appears that the dust-lint mixture made up entirely of long fibers affects the filter performance characteristics in substantially the same manner as a similar dust-lint mixture made up of short and long fibers of the same ratio as found in lint removed from filters. For this reason the long fibers which are more easily prepared should be satisfactory for the comparative rating of filters.

2. The percentage by weight of lint to dust in air has not been definitely determined, and before selecting any fixed standard mixture for rating of filters more study should be given to this point. For those filters, however, which are affected by lint the relative ratings probably would be substantially the same regardless of the percentage of lint used.

3. Lint in air has a dominating influence on the life and dust holding capacity of some filters. With three of the four filters tested the addition of 10 per cent by weight of long fiber lint to the test mixture reduced the life to less than 20 per cent of that determined with no lint in the mixture. Additional lint up to 20 per cent showed relatively little further reduction in life.

4. Lint in air tends to increase the arrestance of most air filters slightly. With all

filters tested increasing percentages of lint by weight resulted in increasing arrestances, partly due to the increased dust eliminating efficiency of the media when containing lint deposits, and partly due to the fact that the filters were practically 100 per cent efficient in removing lint.

5. The effect of lint on the performance characteristics of filters appears to be greater for that type in which the filter capacity depends upon the distribution of dust throughout its depth. For this type of filter the effect of lint is increased when some section is of such construction as to trap the lint and cause abnormally high localized pressure drops.

6. A filter which is designed with large surface area for the purpose of holding the dust on the surface of the filter media may not be seriously affected in its performance characteristics by lint in the air, as there is a large exposed surface area for catching the lint. Furthermore, the lint may be of the same general type of material as used for the filter media, thus having the effect of increasing the thickness of the media without seriously interfering with its performance.

7. Since in practice many filters are called upon to eliminate rather large percentages of lint from the air, and since lint in the air has a dominating influence upon the filter performance characteristics, lint should be considered as an element in the standard dust used for rating air filters.

These tests were conducted for the purpose of studying the characteristics of different types of filters when tested with varying mixtures of dust and lint, and not for the purpose of comparing the different filters used. Any attempts to compare the relative merits of the filters would be misleading. It should not be construed that those filters which showed the greatest reduction in life with increasing percentages of lint were also those with the shortest life in hours, as that was not always the case. The filters used were selected because they were typical of their classifications, and it should not be considered that they are necessarily superior to other filters within their classifications.

The authors wish to acknowledge the cooperation of F. C. Houghten, Director, ASHVE Research Laboratory; and the Technical Advisory Committee on Air Cleaning and Atmospheric Impurities. It is also gratifying to note that the manufacturers of the various types of filters used were generous in the supply of any materials requested, and gave their full cooperation to make the program possible.

DISCUSSION

C. F. MALLY: By the life of the filter, did you mean the time when the filter would be totally opaque to the passage of air? Did that decrease as the different filters filled up?

PROF. A. B. ALGREN: The length of test was governed by the time required to build a resistance rise of 0.25 in. of water across the filter.

A. J. RUMMEL: In the field investigations made what per cent of lint by weight was actually found in office buildings and residences? The results of the investigation are based on certain percentages of lint in the air, but what the actual percentage found in the field in normal cases might be expected was not given.

PROFESSOR ALGREN: This investigation was really a start on the lint study, and I do not believe a definite answer could be made to that question. It is probable that from 10 to 20 per cent of the total weight of dust in the air could be attributed to lint.

G. E. MAY: In feeding that synthetic lint was the same type of feeder used that was used on previous tests and was it not possible for some of that lint to carry the dust on its cells.

PROFESSOR ALGREN: The same equipment was used in this investigation as previously used at the University. The short fibers were mixed with the dust and fed together with the dust. The long fibers were fed separately and intermittently.

P. B. ELLIOTT: By what means was it determined that no lint passed the filter?

PROFESSOR ALGREN: A visual inspection was made of the dust collected in the crucible on the down stream side of the filter.

DYNAMIC AND THERMAL BEHAVIOR OF WATER DROPS IN EVAPORATIVE COOLING PROCESSES

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DROP DYNAMICS

THE dynamic behavior of a water drop as it rises or falls in still or moving air is the introductory phase of the study of the energy transfer phenomenon from sprays to the surrounding gas. The results of preliminary calculations were included in Part 1 of this report.[1]

Falling Drops, Initially at Zero Velocity, in Still Air

Briefly, the equation of motion

$$mg - C\frac{A\rho v^2}{2} = m\frac{dv}{d\theta} . \quad . . . . (1a)$$

is rearranged in a form convenient for graphical integration as follows:

$$\int_0^\theta d\theta = \int_0^v \frac{dv}{g - C\frac{A\rho v^2}{2m}} \quad . . . . (2a)$$

and $$\int_0^S dS = \int_0^\theta v d\theta . \quad . . . . (2b)$$

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** Professor, Mech. Engrg., University of California, Berkeley, Calif. MEMBER ASHVE.
[1] Boelter, L. M. K., ASHVE RESEARCH REPORT No. 1138—Cooling Tower Performance Studies. (ASHVE TRANSACTIONS, Vol. 45, 1939, p. 615.)

for a drop falling from rest into still air. The resistance coefficient has been obtained from several sources [2, 3, 4, 5, 6, 7] and is represented in Fig. 1 for spheres, water drops, methyl salicylate drops (surface tension approximately one-half that of water), and flat plates. Steady-state Reynolds' moduli (Table 2a) which describe the maximum velocity with which drops of various sizes fall through still air, and the limiting Reynolds' moduli corresponding to the maximum velocities for which various drop sizes are stable, are also presented in the same figure, see also Table 2b. The limiting Reynolds' moduli are nearly independent of equivalent drop diameter indicating that the limiting velocities vary approximately inversely as the equivalent diameters. The data of Flower
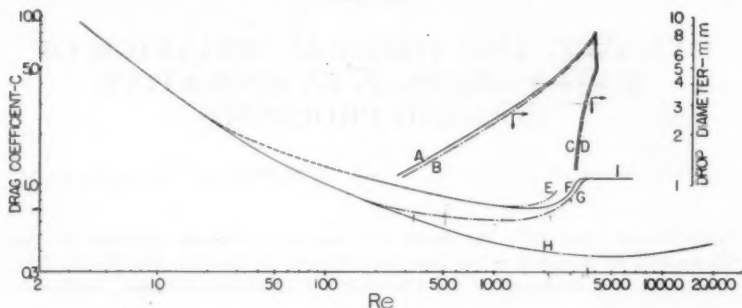
FIG. 1.   DROP DYNAMICS

STEADY STATE REYNOLDS' NUMBER
PLOTTED AS A FUNCTION OF DROP DIAMETER

A   Flower and Wieselsberger
B   Liznar and Wieselsberger

LIMITING REYNOLDS' NUMBER FOR STABLE DROP
AS A FUNCTION OF DROP SIZE

C   Lenard and Flower
D   Lenard and Liznar

DRAG COEFFICIENT AS A FUNCTION OF REYNOLDS' NUMBER

E	Methyl salicylate drop	—Flower
F	Water drops	—Flower
G	Water drops	—Liznar
H	Spheres	—Wieselsberger
I	Flat plates	—Wieselsberger

were eventually thought to be the least reliable of those available and were not used in later computations.

The results of the graphical integrations for the fall of drops from rest in still air are revealed in Figs. 2 and 3 for equivalent drop diameters equal to 0.2, 0.5, 1.0, 2.0, 3.0, 4.0 and 6.0 mm. The resistance data of Flower (F) and Liznar (L) were used to establish the drag coefficient (C). The steady-state velocity curve is almost a linear function of equivalent drop diameter for drops

[2] Liznar, J., *Meteorol. Zeitschr.*, 31, 339 (1914).
[3] Schmidt, W., *Aka. Wiss. Wien Sitzungsber.*, 118:2a, 71 (1909).
[4] Lenard, P., *Meteorol. Zeitschr.*, 21, 249 (1904).
[5] Lenard, P., *Ann. Physik*, IV-65, 629 (1921).
[6] Flower, W. D., *Proc. Phys. Soc. London*, 40, 167 (1928).
[7] Wieselsberger, C., *Zeitschr. Flugtech. Motorluftschiffahrt*, 5, 140 (1914).
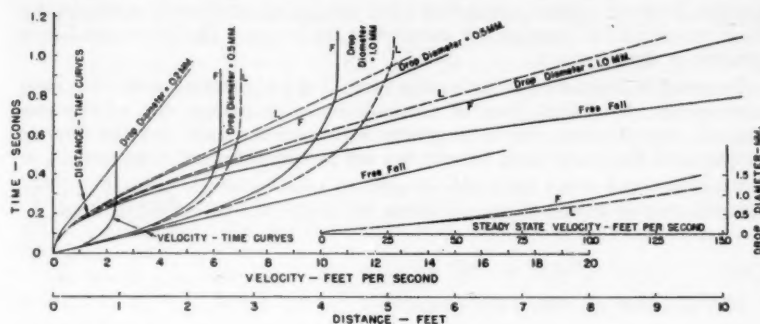
FIG. 2. *A*. SPACE-VELOCITY-TIME CURVES FOR WATER DROPS FALLING FROM REST
THROUGH STILL AIR. *B*. STEADY STATE VELOCITY-EQUIVALENT DIAMETER CURVES

below 1.0 mm in diameter. The maximum steady-state velocity appears to
correspond to a drop diameter of approximately 6 mm. Curves *B* and *D* of
Fig. 1 indicate that the steady-state velocity and the limiting velocity coincide
for a drop diameter of 8 mm; this magnitude defines the maximum stable drop.
Tables 1 and 2 contain the data underlying the curves drawn in Figs. 2 and 3
and Table 3 includes interpolation equations which augment Figs. 2 and 3.

Falling Drops Ejected from Nozzles Into Still Air

Drops ejected vertically downward from nozzles into stationary air and
moving at the limiting velocity, the greatest velocity for which the drop is
stable, will decelerate to the steady-state velocity under the action of the air
resistance. The graphical integration is accomplished by the aid of Equations
2 *a* and *b*, the lower limit of integration being the limiting velocity. The
space-velocity-time results for drop sizes from 0.2 to 6.0 mm equivalent diam-
eters are graphed in Figs. 4, 5 and 6 and tabulated in Tables 4 and 5. Inter-
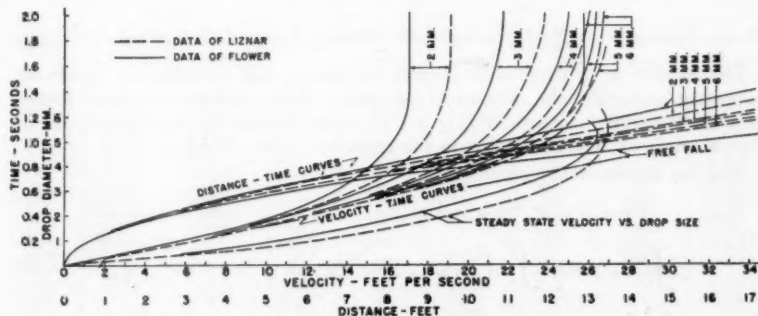
FIG. 3. *A*. SPACE-VELOCITY-TIME CURVES FOR WATER DROPS FALLING FROM REST
THROUGH STILL AIR. *B*. STEADY STATE VELOCITY-EQUIVALENT DIAMETER CURVES

polation between tabular values will yield results of satisfactory accuracy for those magnitudes of distance not included on the curves. The curves are nearly straight in these regions.

For smaller drops the predominating force is the air resistance and the steep curve slopes reveal high rates of deceleration. The smaller rate of deceleration of large drops is due to a greater mass and the fact that the limiting velocity and the steady-state velocity are not greatly different in magnitude.

These results are not applicable directly to cases in which descending drops are coalesced or shattered upon collision but apply only to individual drops.

Drops Projected Upward in Still Air

The equation of motion becomes:

$$mg + C\frac{A\rho v^2}{2} = m\frac{dv}{d\theta} \qquad \qquad \qquad (1b)$$

because the force of gravity and the air resistance both act to decelerate the drop. The limits of integration are the limiting velocity and zero velocity, the latter corresponding to the maximum rise of the drop. Resistance coefficients according to Liznar were used for these computations. The results of the computations are revealed in Figs. 7, 8 and 9, and are tabulated in Tables 6 and 7.

The maximum distance of rise, approximately 16 ft, corresponds to drops of 2 to 3 mm diameter. In this size range the air resistance is least effective in opposing the inertia force during deceleration from the limiting velocity. The maximum height noted for each drop size refers to the trajectory of a single drop. A greater height would be attained by a stream of many drops discharged upward together because the flow system about any single drop is changed by drop interference, collision, and the jet action of the air set into motion by the liquid stream. A decrease in both the relative velocity with respect to the air and the magnitude of the resistance coefficient would result. For the case of jets discharging upward the height of the undispersed jet should be added to the computed height for each drop size to obtain the actual rise above the level of the nozzle.

Drops Falling from Rest Through Air Moving Upward at Various Velocities

The motion of a drop with respect to moving air presents an extremely important problem. The solution of one phase of this problem for drops falling through vertically upward moving air is made possible by the application of the principle of relative motion to the curves of Figs. 2 and 3.

For the system under consideration

$$\left.\begin{array}{l} V_{drop \rightarrow air} = V_{drop} + V_{air} \\[2mm] \int_0^\theta V_{drop}\, d\theta = \int_0^\theta \left(V_{drop \rightarrow air} - V_{air} \right) d\theta \\[2mm] S_{drop} = S_{Figs.\ 2\ and\ 3} - V_{air}\,\theta \end{array}\right\} \qquad \qquad (3)$$

The velocities plotted in Figs. 2 and 3 are those of the drop relative to that

of the air ($V_{drop \rightarrow air}$) through which it is traveling. $S_{Figs.\ 2\ and\ 3}$ is a function of time (θ) and, when known, $V_{drop \rightarrow air}$ is fixed.

For a given displacement of the drop in space a graphical solution of Equation (3) may be employed. A straight edge is placed on the curve sheet with a slope $1/V_{air}$ and intersecting the point ($S = S_{drop}$, $\theta = O$). Intersection of the line thus defined with the distance-time curve yields the time necessary to traverse the distance through the moving air stream. The corresponding relative velocity of the drop ($V_{drop \rightarrow air}$) is read from the velocity-time curve and the absolute velocity of the drop (V_{drop}) is computed from the first of Equations (3). The resistance data of Liznar were used for these computations.

The results for distances of fall from 0.5 to 15 ft and air stream velocities upward of 2, 4, 6 and 10 fps are plotted in Figs. 10 to 14 and tabulated in

TABLE 1—VELOCITY-TIME DATA FOR WATER DROPS FALLING FROM REST THROUGH STILL AIR

DROP DIAMETER = 0.2MM		DROP DIAMETER = 0.5MM			DROP DIAMETER = 1.0MM		
	θ, sec		θ, sec			θ, sec	
v ft/sec	(Flower and Liznar)	v ft/sec	(Flower)	(Liznar)	v ft/sec	(Flower)	(Liznar)
0.5	0.0168	1.0	0.0328	0.0328	1.0	0.0311	0.0311
1.0	0.0372	2.0	0.0680	0.0670	2.0	0.0642	0.0641
1.5	0.0646	3.0	0.1082	0.1060	3.0	0.0997	0.0986
2.0	0.1210	4.0	0.1612	0.1545	4.5	0.1558	0.1531
2.25	0.1783	5.0	0.2357	0.2188	6.0	0.2230	0.2158
2.367*	0.240	5.5	0.2995	0.2615	7.5	0.3098	0.2858
		6.0	0.4135	0.3244	8.5	0.3910	0.3415
		6.25*	0.580	0.3668	9.0	0.4442	0.3756
		6.50	...	0.4207	9.5	0.522	0.4138
		6.75	...	0.510	10.0	0.634	0.4542
		7.02*	...	0.680	10.63*	0.840	...
					11.0	...	0.565
					12.0	...	0.7685
					12.72*	...	1.010

DROP DIAMETER = 2.0MM			DROP DIAMETER = 3.0MM			DROP DIAMETER = 4.0MM		
	θ, sec			θ, sec			θ, sec	
v ft/sec	(Flower)	(Liznar)	v ft/sec	(Flower)	(Liznar)	v ft/sec	(Flower)	(Liznar)
1.5	0.0466	0.0466	2.0	0.0622	0.0622	2.0	0.0622	0.0622
3.0	0.0940	0.0940	4.0	0.1268	0.1268	4.0	0.125	0.125
4.5	0.146	0.146	6.0	0.191	0.1910	6.0	0.191	0.189
6.0	0.1980	0.1978	8.0	0.266	0.2654	8.0	0.258	0.256
7.5	0.255	0.251	10.0	0.342	0.330	10.0	0.332	0.329
9.0	0.3176	0.3064	12.0	0.428	0.409	12.0	0.410	0.404
10.5	0.390	0.370	14.0	0.554	0.507	14.0	0.497	0.485
12.0	0.4804	0.446	17.0	0.741	0.668	17.0	0.651	0.626
14.0	0.6484	0.566	19.0	0.953	0.816	20.0	0.858	0.802
15.0	0.7676	0.642	20.0	1.104	...	22.0	1.061	0.961
16.0	0.9636	0.733	20.9	1.365	...	23.0	1.222	1.071
16.4	1.1004	...	21.0	...	1.038	24.1	1.521	1.237
17.0	...	0.8516	21.8*	1.920	...	25.02*	2.100	...
17.13*	1.50	...	22.0	...	1.208	25.2	...	1.535
18.0	...	1.0472	22.9	...	1.466	26.03*	...	1.920
18.4	...	1.191	23.83*	...	1.900			
19.2*	...	1.66						

* Steady-state velocity.

TABLE 1—VELOCITY-TIME DATA FOR WATER DROPS FALLING FROM REST THROUGH STILL AIR (Continued)

DROP DIAMETER = 5.0MM			DROP DIAMETER = 6.0MM			FREE FALL	
v ft/sec	θ, sec		v ft/sec	θ, sec		v ft/sec	θ, sec
	(Flower)	(Liznar)		(Flower)	(Liznar)		
2.5	0.0778	0.0778	2.5	0.0778	0.0778	2.0	0.0622
5.0	0.1576	0.1576	5.0	0.1578	0.1578	4.0	0.1244
7.5	0.238	0.238	7.5	0.238	0.238	6.0	0.1866
10.0	0.3256	0.3234	10.0	0.3234	0.322	8.0	0.2488
12.5	0.418	0.415	12.5	0.413	0.412	10.0	0.311
15.0	0.522	0.515	15.0	0.511	0.507	12.0	0.373
19.0	0.716	0.697	19.0	0.690	0.677	14.0	0.435
22.0	0.921	0.880	22.0	0.871	0.846	16.0	0.4975
24.0	1.127	1.064	23.5	1.008	0.957	18.0	0.560
25.0	1.308	1.204	24.5	1.120	1.058	20.0	0.622
25.5	1.455	...	25.1	1.308	...	22.0	0.684
26.1	...	1.460	25.5	...	1.226	24.0	0.747
26.4[a]	1.880	...	25.8[a]	1.86	...	26.0	0.808
26.9[a]	...	1.850	26.1	...	1.389	28.0	0.870
			26.8[a]	...	2.00	30.0	0.933
						32.0	0.995
						34.0	1.058
						36.0	1.119
						38.0	1.181
						40.0	1.243

[a] Steady-state velocity.

Table 8. The high relative velocity between the drop and the air stream increases the rates of heat and mass transfer, but the loss of small drops (called drift) offsets advantages realized beyond a certain optimum condition which obtains for each particular instance. Fig. 15 reveals the diameters of drops, all sizes below which may constitute drift in a contraflow tower. The relative velocity affects the rate of heat and mass transfer, but the absolute velocity is the determining factor in fixing the size of the equipment and in the impact process when a drop strikes wetted solid surfaces.

Computations of the Evaporative Cooling of Water Drops in Air

The decrease in temperature of a drop moving with respect to air may be computed by integrating equations which express an energy balance and a mass balance. The idealized water-air system may be described as follows:

1. Distance-velocity-time data included in this paper describe the behavior of the drops.
2. The relation between unit mass conductance and unit thermal conductance as determined from psychrometric data is valid.
3. The transient problem may be analyzed as a succession of steady-state conditions.
4. During the period of acceleration (positive and negative) the drops are spheres of infinite thermal conductivity.
5. Unit thermal conductances in the range available for spheres [a] also apply to the water drops,

$$Nu = 0.70\, Re^{0.55}, \quad 10^3 < Re < 10^5 \quad \ldots \quad \ldots \quad \ldots \quad (4)$$

[a] Büttner, K., *Veröffentlichungen des Preussischen Meteorologischen Institutes, Abhandlung*, 10, No. 5 (1934).

6. Unit thermal conductances for spheres in the range not available from the literature are obtained by the application of the proportion:

$$\frac{(Nu_{Re \, < \, 10^2}) \text{ spheres}}{(Nu_{Re \, < \, 10^2}) \text{ cylinders}} = \frac{0.70 \, Re^{0.52}}{0.35 \, Re^{0.56}} \quad \cdots \quad \cdots \quad (5)$$

where[9]: $Nu = 0.35 \, Re^{0.56}, \; 10^2 < Re < 10^6$ . . . . (6)

represents the thermal behavior of cylinders in cross flow. The data for cylinders at small Reynolds' moduli are obtained from McAdams.[10] The magnitudes employed in the computations are revealed in Fig. 16.

7. The equivalent radius (R_o) of the drop is constant during the evaporation process.

TABLE 2—DISTANCE-TIME DATA FOR WATER DROPS FALLING FROM REST THROUGH STILL AIR

DROP DIAMETER = 0.2MM		DROP DIAMETER = 0.5MM			DROP DIAMETER = 1.0MM		
θ sec	S, ft	θ sec	S, ft		θ sec	S, ft	
	(Flower and Liznar)		(Flower)	(Liznar)		(Flower)	(Liznar)
0.07	0.0623	0.10	0.1468	0.1482	0.10	0.157	0.157
0.12	0.1516	0.16	0.355	0.361	0.18	0.481	0.486
0.18	0.280	0.24	0.718	0.737	0.24	0.824	0.842
0.24 [a]	0.429	0.30	1.038	1.077	0.30	1.241	1.275
0.44	0.893	0.38	1.495	1.560	0.38	1.875	1.953
0.64	1.367	0.46	1.981	2.087	0.46	2.58	2.71
0.84	1.840	0.52	2.359	2.489	0.54	3.33	3.546
		0.58 [a]	2.733	. . .	0.62	4.11	4.43
		0.60	. . .	3.038	0.72	5.13	5.59
		0.68 [a]	. . .	3.596	0.82	. . .	6.79
		0.88	4.608	. . .	0.84 [a]	6.39	. . .
		0.98	. . .	5.702	0.92	. . .	8.03
					1.01 [a]	. . .	9.17
					1.07	. . .	9.93
					1.14	9.58	. . .

DROP DIAMETER = 2.0MM		DROP DIAMETER = 3.0MM			DROP DIAMETER = 4.0MM		
θ sec	S, ft	θ sec	S, ft		θ sec	S, ft	
	(Flower)	(Liznar)		(Flower)	(Liznar)		(Flower)	(Liznar)
0.10	0.159	0.159	0.10	0.160	0.160	0.20	0.638	0.638
0.20	0.616	0.616	0.20	0.630	0.630	0.28	1.23	1.23
0.28	1.18	1.20	0.28	1.21	1.21	0.36	2.008	2.020
0.36	1.906	1.934	0.36	2.000	2.014	0.42	2.71	2.73
0.42	2.54	2.60	0.42	2.68	2.72	0.48	3.476	3.514
0.48	3.242	3.326	0.48	3.392	3.488	0.56	4.64	4.71
0.56	4.22	4.37	0.56	4.48	4.63	0.64	5.916	6.012
0.64	5.32	5.54	0.64	5.644	5.892	0.72	7.31	7.48
0.72	6.48	6.79	0.72	6.94	7.25	0.80	8.818	9.104
0.80	7.64	8.08	0.80	8.32	8.72	0.90	10.82	11.09
0.90	9.22	9.82	0.90	10.12	10.64	1.00	12.914	13.260
1.00	10.80	11.54	1.00	12.02	12.70	1.20	17.352	17.882
1.20	14.08	15.18	1.20	16.04	17.00	1.40	22.02	22.74
1.40	17.46	18.90	1.40	20.18	21.46	1.60	26.80	27.76
1.50 [a]	19.16	. . .	1.60	24.42	26.06	1.80	31.68	32.90
1.60	. . .	22.70	1.80	28.72	30.76	1.92 [a]	. . .	36.02
1.66 [a]	. . .	23.84	1.90 [a]	. . .	33.14	2.00	36.62	. . .
			1.92 [a]	31.32	. . .	2.10 [a]	39.12	. . .

[a] Steady-state attained.

[9] Reiher, H., *V.d.I. Forschungsarbeiten*, No. 322 (1929).
[10] McAdams, W. H., Heat Transmission, p. 219, McGraw-Hill Book Co., New York, 1933.

TABLE 2—DISTANCE-TIME DATA FOR WATER DROPS FALLING FROM REST THROUGH STILL AIR (Continued)

	DROP DIAMETER = 5.0MM			DROP DIAMETER = 6.0MM			FREE FALL
	S, ft				S, ft			S, ft
θ sec	(Flower)	(Liznar)	θ sec	(Flower)	(Liznar)	θ sec	(Flower and Liznar)
0.20	0.641	0.641	0.20	0.643	0.643	0.05	0.0402
0.28	1.24	1.24	0.28	1.25	1.25	0.1	0.1608
0.36	2.018	2.022	0.36	2.036	2.036	0.2	0.6432
0.42	2.74	2.75	0.42	2.76	2.76	0.3	1.4472
0.48	3.538	3.550	0.48	3.554	3.558	0.4	2.5728
0.56	4.725	4.76	0.56	4.77	4.78	0.5	4.020
0.64	6.064	6.108	0.64	6.128	6.158	0.6	5.789
0.72	7.55	7.63	0.72	7.66	7.74	0.7	7.879
0.80	9.12	9.22	0.80	9.26	9.34	0.8	10.291
0.90	11.25	11.38	0.90	11.44	11.58	0.9	13.025
1.00	13.46	13.66	1.00	13.74	13.9u	1.0	16.080
1.20	18.22	18.51	1.20	18.60	18.86	1.1	19.457
1.40	23.20	23.66	1.40	23.64	24.00	1.2	23.155
1.60	28.34	28.90	1.60	28.74	29.24	1.3	27.175
1.80	33.56	34.24	1.80	33.88	34.54	1.4	31.417
1.85*	...	35.58	1.86*	35.42	...	1.5	36.180
1.88	35.66	...	2.00*	...	39.90	1.6	41.165
						1.7	46.471
						1.8	52.099
						1.9	58.040
						2.0	64.320

* Steady-state attained.

TABLE 2a—STEADY-STATE DATA FOR DROPS FALLING IN AIR

DROP DIAMETER d, MM	LIZNAR (WATER)			FLOWER (WATER)			FLOWER (METHYL SALICYLATE)		
	Resistance Coefficient C_{s-s}	Reynolds Number Re_{s-s}	Velocity V_{s-s} ft/sec	Resistance Coefficient C_{s-s}	Reynolds Number Re_{s-s}	Velocity V_{s-s} ft/sec	Resistance Coefficient C_{s-s}	Reynolds Number Re_{s-s}	Velocity V_{s-s} ft/sec
0.05	...	0.183	0.18*	...	0.183	0.18*	...	0.183	0.18*
0.2	4.18	9.64	2.367	4.18	9.64	2.367	4.18	9.64	2.367
0.5	1.19	71.4	7.02	1.50	63.6	6.25	1.50	63.6	6.25
1.0	0.725	259.0	12.72	1.04	216.0	10.63	0.999	241.0	11.8
2.0	0.635	782.0	19.21	0.802	698.0	17.13	0.795	763.0	18.8
3.0	0.621	1455.0	23.83	0.740	1335.0	21.80	0.759	1432.0	23.5
4.0	0.690	2120.0	26.03	0.758	2035.0	25.02	0.829	2110.0	26.0
5.0	0.807	2740.0	26.9	0.845	2690.0	26.4	0.98	2490.0	24.5
6.0	0.98	3275.0	26.8	1.06	3150.0	25.8	...	...	...

* Computed by Stokes' Law.　s-s refers to steady state.

TABLE 2b—LIMITING VELOCITIES FOR STABLE WATER DROPS

DROP DIAMETER MM	VELOCITY INCREMENT ft/sec	LIMITING VELOCITY FT/SEC		LIMITING REYNOLDS NUMBER	
		(Flower)	(Liznar)	(Flower)	(Liznar)
0.2	*	690.0	735.0	...	...
0.5	*	285.0	295.0	2900	3000
1.0	137.	147.0	152.0	3050	3150
2.0	62.6	79.7	81.0	3250	3300
3.0	35.2	57.0	58.9	3470	3550
4.0	21.0	46.0	47.0	3740	3830
5.0	13.1	39.5	40.0	4030	4070
6.0	6.6	32.4	33.4	4060	4075

* Note: Limiting velocity determined by extrapolation.
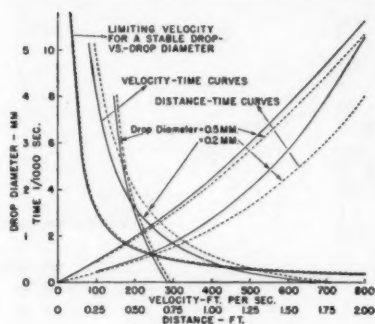
FIG. 4.   A. SPACE-VELOCITY-TIME CURVES
FOR DROPS PROJECTED DOWNWARD AT
THE LIMITING VELOCITY INTO STILL AIR
AND DECELERATED TO THE STEADY-STATE
VELOCITY
(TIME RANGE: $0 < \theta < 0.01$ SEC.)
B. LIMITING VELOCITY OF A STABLE
DROP AS A FUNCTION OF EQUIVALENT
DROP DIAMETER
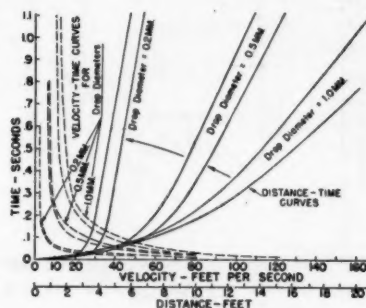
FIG. 5.   SPACE-VELOCITY-TIME CURVES FOR
DROPS PROJECTED DOWNWARD AT THE
LIMITING VELOCITY INTO STILL AIR AND
DECELERATED   TO   THE   STEADY-STATE
VELOCITY
(TIME RANGE: $0 < \theta < 1.0$ SEC.)

In cooling tower work the radius decreases in magnitude, but not over 5 per cent.
    8. Drop diameters are not changed by collision.
    The application of an energy balance and a mass balance (in time interval $d\theta$) to
the ideal system described leads to the following equation:

$$\frac{3}{2}(Nu)\, d\, (Fo)\left(1 + \frac{f_r}{f_c}\right) = \frac{\dfrac{dt}{t - t_a}}{\left(\dfrac{k_a}{k_1}\right)\left[1 + \dfrac{U'_r}{R_v T_m f_c \left(1 + \dfrac{f_r}{f_c}\right)}\left(\dfrac{P_w - P_g}{t - t_a}\right)\right]} \quad \cdots \quad (7)$$
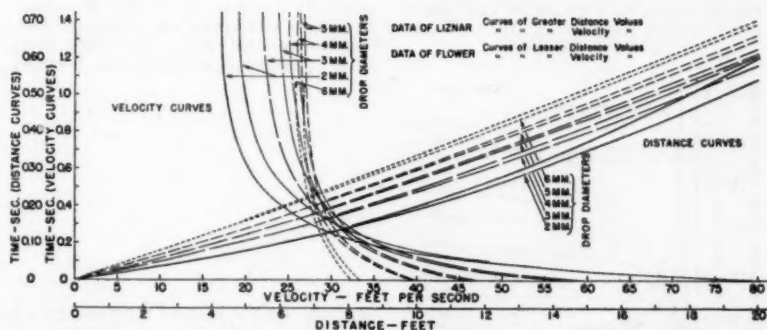
FIG. 6.   SPACE-VELOCITY-TIME CURVES FOR DROPS PROJECTED DOWNWARD AT THE
LIMITING VELOCITY INTO STILL AIR AND DECELERATED TO THE STEADY-STATE VELOCITY
(TIME RANGE: $0 < \theta < 1.00$ SEC.)
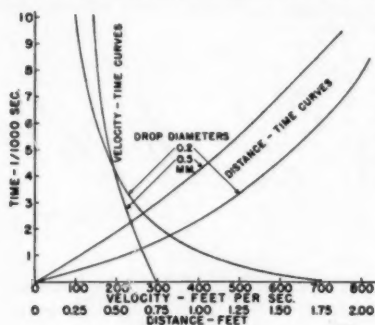
FIG. 7.   SPACE-VELOCITY-TIME CURVES FOR
DROPS PROJECTED UPWARD AT THE LIMIT-
ING VELOCITY INTO STILL AIR. RESISTANCE
COEFFICIENTS   FROM   LIZNAR.   (TIME:
$0 < \theta < 0.010$ SEC.)
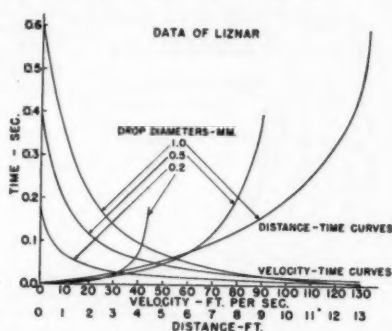
FIG. 8.   SPACE-VELOCITY-TIME CURVES FOR
DROPS PROJECTED UPWARD AT THE LIMIT-
ING   VELOCITY   INTO   STILL   AIR.   (TIME:
$0 < \theta < 0.6$ SEC.)
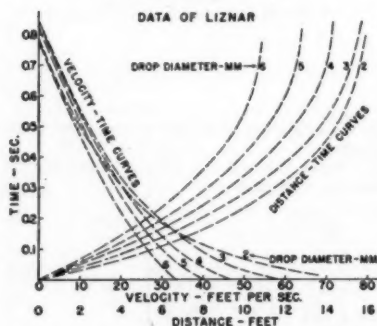
FIG. 9.   SPACE-VELOCITY-TIME CURVES FOR
DROPS PROJECTED UPWARD AT THE LIMIT-
ING VELOCITY INTO STILL AIR. (TIME:
$0 < \theta < 0.6$ SEC.)
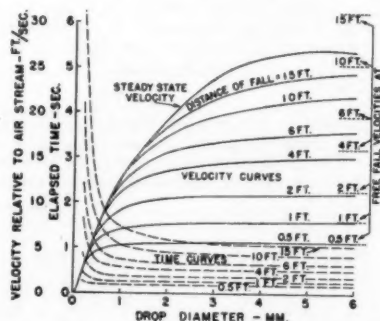
FIG. 10.   DROP VELOCITY RELATIVE TO
AIR AND ELAPSED TIME AS A FUNCTION
OF DROP DIAMETER FOR VARIOUS DIS-
TANCES OF FALL FROM REST INTO STILL
AIR.   (RESISTANCE DATA FROM LIZNAR)
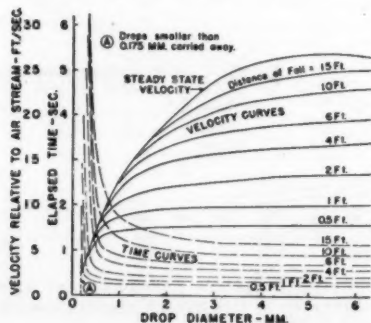
FIG. 11.   DROP VELOCITY RELATIVE TO AIR
AND ELAPSED TIME AS A FUNCTION OF
DROP DIAMETER FOR VARIOUS DISTANCES
OF FALL FROM REST INTO AIR MOVING
UPWARD AT 6 FPS
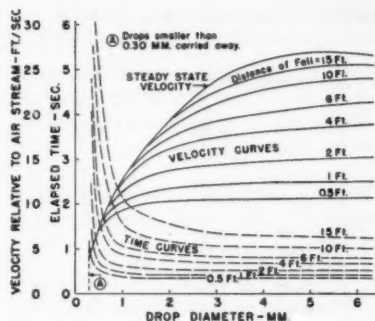
FIG. 12. DROP VELOCITY RELAT.   TO AIR AND ELAPSED TIME AS A FUNCTION OF DROP DIAMETER FOR VARIOUS DISTANCES OF FALL FROM REST INTO AIR MOVING UPWARD AT 2 FPS
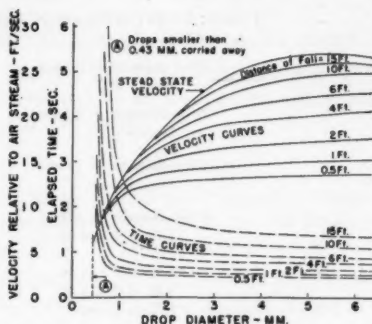
FIG. 13. DROP VELOCITY RELATIVE TO AIR AND ELAPSED TIME AS A FUNCTION OF DROP DIAMETER FOR VARIOUS DISTANCES OF FALL FROM REST INTO AIR MOVING UPWARD AT 4 FPS
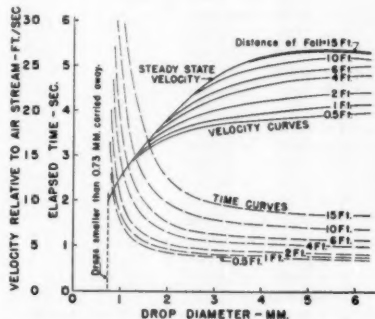
FIG. 14. DROP VELOCITY RELATIVE TO AIR AND ELAPSED TIME AS A FUNCTION OF DROP DIAMETER FOR VARIOUS DISTANCES OF FALL FROM REST INTO AIR MOVING UPWARD AT 10 FPS
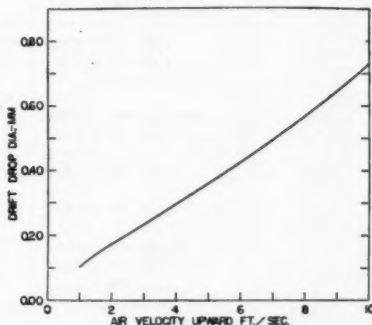
FIG. 15. THE UPPER LIMIT OF DROP SIZES WHICH CONSTITUTE DRIFT IN A CONTRAFLOW TOWER AS A FUNCTION OF UPWARD AIR VELOCITY
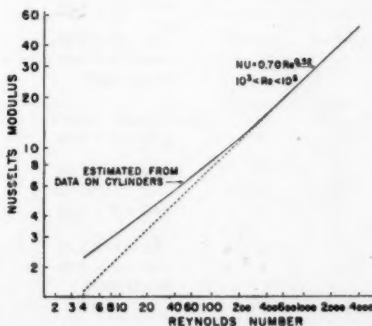
FIG. 16. HEAT TRANSFER FUNCTION FOR SPHERES IN CURRENTS OF AIR

TABLE 3—INTERPOLATION EQUATIONS AUGMENTING FIGS. 2 AND 3

Drop diameter = 0.2mm
  Steady state. $S = 2.367\theta - 0.148$ [a, b]

Drop diameter = 0.5mm
  Steady state. $S = 6.25\theta - 0.892$ [a]
                 $S = 7.02\theta - 1.174$ [b]

Drop diameter = 1.0mm
  Steady state. $S = 10.63\theta - 2.54$ [a]
                 $S = 12.72\theta - 3.68$ [b]

Drop diameter = 2.0mm
  $1.20 < \theta < 1.40$ $S = 16.9\theta - 6.20$ [a]
                          $S = 18.6\theta - 7.14$ [b]
  $1.40 < \theta < 1.50$ $S = 17.0\theta - 6.34$ [a]
  $1.40 < \theta < 1.66$ $S = 19.0\theta - 7.70$ [b]
  Steady state. $S = 17.13\theta - 6.54$ [a]
                 $S = 19.2\theta - 8.03$ [b]

Drop diameter = 3.0mm
  $1.20 < \theta < 1.40$ $S = 20.7\theta - 8.80$ [a]
                          $S = 22.3\theta - 9.76$ [b]
  $1.40 < \theta < 1.60$ $S = 21.4\theta - 9.78$ [a]
                          $S = 23.0\theta - 10.74$ [b]
  $1.60 < \theta < 1.80$ $S = 21.5\theta - 9.98$ [a]
                          $S = 23.5\theta - 11.54$ [b]
  $1.80 < \theta < 1.90$ $S = 23.8\theta - 12.08$ [b]
  $1.80 < \theta < 1.92$ $S = 21.67\theta - 10.29$ [a]
  Steady state. $S = 23.83\theta - 12.14$ [b]
                 $S = 21.8\theta - 10.54$ [a]

Drop diameter = 4.0mm
  $1.20 < \theta < 1.40$ $S = 23.34\theta - 10.66$ [a]
                          $S = 24.3\theta - 11.28$ [b]
  $1.40 < \theta < 1.60$ $S = 23.9\theta - 11.44$ [a]
                          $S = 25.1\theta - 12.40$ [b]
  $1.60 < \theta < 1.80$ $S = 24.4\theta - 12.24$ [a]
                          $S = 25.7\theta - 13.36$ [b]
  $1.80 < \theta < 1.92$ $S = 26.0\theta - 13.90$ [a]
  $1.80 < \theta < 2.00$ $S = 24.7\theta - 12.78$ [b]
  $2.00 < \theta < 2.10$ $S = 25.0\theta - 13.38$ [a]
  Steady state. $S = 26.03\theta - 13.96$ [b]
                 $S = 25.02\theta - 13.42$ [a]

Drop diameter = 5.0mm
  $1.20 < \theta < 1.40$ $S = 24.9\theta - 11.66$ [a]
                          $S = 25.74\theta - 12.38$ [b]
  $1.40 < \theta < 1.60$ $S = 25.7\theta - 12.78$ [a]
                          $S = 26.2\theta - 13.02$ [b]
  $1.60 < \theta < 1.80$ $S = 26.1\theta - 13.42$ [a]
                          $S = 26.7\theta - 13.82$ [b]
  $1.80 < \theta < 1.88$ $S = 26.3\theta - 13.78$ [a]
  $1.80 < \theta < 1.85$ $S = 26.8\theta - 14.00$ [b]
  Steady state. $S = 26.4\theta - 13.97$ [a]
                 $S = 26.9\theta - 14.19$ [b]

Drop diameter = 6.0mm
  $1.20 < \theta < 1.40$ $S = 25.2\theta - 11.64$ [a]
                          $S = 25.7\theta - 11.98$ [b]
  $1.40 < \theta < 1.60$ $S = 25.5\theta - 12.06$ [a]
                          $S = 26.2\theta - 12.68$ [b]
  $1.60 < \theta < 1.80$ $S = 25.7\theta - 12.28$ [a]
                          $S = 26.5\theta - 13.16$ [b]
  $1.80 < \theta < 1.86$ $S = 25.67\theta - 12.33$ [a]
  $1.80 < \theta < 2.00$ $S = 26.75\theta - 13.60$ [b]
  Steady state. $S = 25.8\theta - 12.57$ [a]
                 $S = 26.8\theta - 13.70$ [b]

[a] Flower.   [b] Liznar.

Except for the term $\left(1 + \dfrac{f_r}{f_c}\right)$, the left side of Equation (7) depends upon dynamical conditions only (since Nu is a function of Re, see Fig. 16) and the right side depends only upon thermal conditions for a given magnitude of the partial pressure of the water vapor far away.

TABLE 4—VELOCITY-TIME DATA FOR WATER DROPS PROJECTED VERTICALLY DOWNWARD THROUGH STILL AIR. DECELERATION FROM THE LIMITING TO THE STEADY-STATE VELOCITY

DROP DIAMETER = 0.2MM			DROP DIAMETER = 0.5MM			DROP DIAMETER = 1.0MM		
v ft/sec	θ, sec		v ft/sec	θ, sec		v ft/sec	θ, sec	
	(Flower)	(Liznar)		(Flower)	(Liznar)		(Flower)	(Liznar)
735.0	...	0	295.0	...	0	152.0	...	0
690.0	0	...	285.0	0	...	147.0	0	...
600.0	0.000170	0.000260	240.0	0.001425	0.001780	125.0	0.00493	0.00628
550.0	0.000270	0.000415	190.0	0.004120	0.004660	100.0	0.01428	0.01674
500.0	0.000396	0.000609	165.0	0.00543	0.00614	90.0	0.0188	0.0220
450.0	0.000500	0.000835	140.0	0.00873	0.010035	80.0	0.0267	0.0309
400.0	0.000952	0.001165	125.0	0.0106	0.01220	70.0	0.0363	0.0424
350.0	0.001235	0.001570	110.0	0.01352	0.01575	60.0	0.0479	0.05667
300.0	0.001667	0.00208	95.0	0.0175	0.0204	50.0	0.0651	0.0770
260.0	0.00216	0.00266	85.0	...	0.02387	42.0	0.0843	0.1019
220.0	0.00281	0.00349	80.0	0.02150	...	32.0	0.1203	0.1487
180.0	0.00375	0.00464	60.0	0.03095	0.03506	24.0	0.173	0.2070
140.0	0.00522	0.00644	50.0	0.0380	0.0452	18.0	0.2515	0.3251
100.0	0.00806	0.00969	42.0	0.04665	0.05062	15.0	0.3251	0.5411
60.0	0.01272	0.01689	35.0	0.0588	0.0725	13.5	...	0.7251
40.0	0.0204	0.0275	28.0	0.0743	0.0938	13.0	0.4091	...
25.0	0.0279	0.0368	20.0	0.0983	0.1286	12.72 a	...	0.96
15.0	0.0359	0.0448	14.0	0.1455	0.1766	11.3	0.6395	...
10.0	0.0539	0.0628	10.0	0.2151	0.2398	10.63 a	0.93	...
7.0	0.0755	0.0844	8.6	...	0.3294			
5.0	0.1043	0.1132	8.0	0.2923	...			
3.6	0.1359	0.1448	7.4	...	0.5022			
2.6	0.2339	0.2428	7.02 a	...	0.660			
2.367 a	0.320	0.340	6.6	0.4655	...			
			6.25 a	0.62	...			

DROP DIAMETER = 2.0MM			DROP DIAMETER = 3.0MM			DROP DIAMETER = 4.0MM		
v ft/sec	θ, sec		v ft/sec	θ, sec		v ft/sec	θ, sec	
	(Flower)	(Liznar)		(Flower)	(Liznar)		(Flower)	(Liznar)
81.0	...	0	58.9	...	0	47.0	...	0
79.7	0	...	57.0	0	...	46.0	0	...
60.0	0.0472	0.0496	50.0	0.0323	0.0378	42.0	0.0390	0.0475
45.0	0.1184	0.1220	43.0	0.0832	0.0900	36.0	0.1292	0.1420
34.0	0.1976	0.2296	36.0	0.1676	0.1854	31.0	0.2848	0.3174
26.0	0.3420	0.4408	31.0	0.281	0.322	29.0	...	0.462
23.0	...	0.5876	27.5	...	0.518	28.5	0.4532	...
22.0	0.4904	...	27.0	0.4524	...	28.0	...	0.5886
21.0	...	0.7960	25.5	...	0.772	27.0	0.6452	0.818
20.0	...	1.006	25.0	0.6126	...	26.03 a	...	1.18
19.2 a	...	1.28	24.5	...	1.042	26.0	0.896	...
19.0	0.7416	...	23.83 a	...	1.38	25.02 a	1.31	...
18.0	0.9300	...	23.5	0.8106	...			
17.13 a	1.24	...	22.5	1.120	...			
			21.8 a	1.40	...			

a Steady-state velocity.

TABLE 4—VELOCITY-TIME DATA FOR WATER DROPS PROJECTED VERTICALLY DOWN-
WARD THROUGH STILL AIR. DECELERATION FROM THE LIMITING TO THE STEADY-
STATE VELOCITY (Continued)

DROP DIAMETER = 5.0MM			DROP DIAMETER = 6.0MM		
v ft/sec	θ, sec		v ft/sec	θ, sec	
	(Flower)	(Liznar)		(Flower)	(Liznar)
40.0	...	0	33.4	...	0
39.5	0	...	32.4	0	...
36.0	0.0656	0.0730	31.0	0.0770	0.1212
32.0	0.1716	0.1892	29.5	0.1832	0.2312
29.5	0.2934	0.3518	28.0	0.3460	0.4054
28.0	0.4422	0.586	27.4	...	0.548
27.5	...	0.780	27.0	0.520	...
27.0	0.6812	...	26.8 [a]	...	0.77
26.9 [a]	...	1.11	26.4	0.680	...
26.4 [a]	1.22	...	25.8 [a]	0.91	...

[a] Steady-state velocity.

Dynamical Function

To evaluate

$$\frac{3}{2} \cdot Nu \cdot d(Fo) \cdot \left(1 + \frac{f_r}{f_c}\right) = \psi d\theta \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

the following procedure is utilized: reference to the velocity-time curve yields
Re, then Nu is obtained from Fig. 16, and the magnitude of f_c computed from

TABLE 5—DISTANCE-TIME DATA FOR WATER DROPS PROJECTED VERTICALLY DOWN-
WARD THROUGH STILL AIR. DECELERATION FROM THE LIMITING TO THE STEADY-
STATE VELOCITY

DROP DIAMETER = 0.2MM			DROP DIAMETER = 0.5MM			DROP DIAMETER = 1.0MM		
θ sec	S, ft		θ sec	S, ft		θ sec	S, ft	
	(Flower)	(Liznar)		(Flower)	(Liznar)		(Flower)	(Liznar)
0.0012	0.5273	0.6229	0.0012	0.3186	0.3290	0.01	1.31	1.36
0.0024	0.9250	1.0204	0.0024	0.5296	0.6158	0.02	2.30	2.39
0.0036	1.174	1.316	0.0036	0.8319	0.8647	0.04	3.83	4.02
0.0048	1.368	1.540	0.0048	1.0480	1.0880	0.07	5.50	5.86
0.0060	1.538	1.733	0.0060	1.246	1.296	0.10	6.77	7.29
0.0076	1.716	1.951	0.0076	1.494	1.550	0.14	8.06	8.82
0.0100	1.943	2.214	0.0100	1.829	1.904	0.18	9.10	10.02
0.020	2.495	2.937	0.020	2.96	3.08	0.24	10.32	11.42
0.040	2.954	3.649	0.040	4.21	4.54	0.30	11.40	12.67
0.070	3.249	4.00	0.070	5.35	5.86	0.38	12.58	14.12
0.100	3.432	4.22	0.100	6.06	6.80	0.46	13.60	15.46
0.14	3.58	4.43	0.14	6.74	7.68	0.54	14.56	16.70
0.18	3.71	4.57	0.18	7.24	8.30	0.62	15.49	17.87
0.24	3.87	4.71	0.24	7.82	8.97	0.70	16.39	18.93
0.32 [a]	4.07	...	0.30	8.33	9.53	0.78	17.28	20.05
0.34 [a]	...	4.94	0.38	8.93	10.20	0.86	18.15	21.09
0.50	4.50	5.32	0.46	9.51	10.84	0.93 [a]	18.90	...
1.00	5.68	6.50	0.54	10.03	11.44	0.96 [a]	...	22.11
			0.62 [a]	10.54	...	1.10	20.71	23.89
			0.66 [a]	...	12.30			
			0.80	11.67	13.28			
			1.00	12.92	14.69			

[a] Steady-state attained.

TABLE 5—DISTANCE-TIME DATA FOR WATER DROPS PROJECTED VERTICALLY DOWN-WARD THROUGH STILL AIR. DECELERATION FROM THE LIMITING TO THE STEADY-STATE VELOCITY (Continued)

	DROP DIAMETER = 2.0MM			DROP DIAMETER = 3.0MM			DROP DIAMETER = 4.0MM	
θ sec	S, ft		θ sec	S, ft		θ sec	S, ft	
	(Flower)	(Liznar)		(Flower)	(Liznar)		(Flower)	(Liznar)
0.01			0.01			0.01		
0.02	1.477	1.504	0.02	1.096	1.129	0.02	0.894	0.932
0.04	2.784	2.832	0.04	2.108	2.164	0.04	1.746	1.780
0.07	4.530	4.602	0.07	3.502	3.582	0.07	2.970	3.018
0.10	6.06	6.156	0.10	4.788	4.896	0.10	4.126	4.198
0.15	8.25	8.378	0.15	6.76	6.90	0.15	5.940	6.048
0.20	10.09	10.30	0.20	8.54	8.73	0.20	7.624	7.760
0.25	11.68	12.10	0.25	10.18	10.44	0.25	9.184	9.412
0.30	13.10	13.60	0.30	11.74	12.06	0.30	10.802	10.998
0.35	14.42	15.08	0.35	13.24	13.60	0.36	12.613	12.843
0.40	15.68	16.48	0.40	14.66	15.07	0.42	14.37	14.64
0.48	17.53	18.56	0.48	16.84	17.36	0.50	16.65	16.96
0.56	19.25	20.49	0.56	18.92	19.55	0.60	19.41	19.79
0.66	21.28	22.77	0.66	21.43	22.22	0.70	22.11	22.55
0.76	23.21	25.07	0.76	23.84	24.80	0.80	24.76	25.28
0.86	25.06	27.03	0.86	26.19	27.33	0.90	27.37	27.97
0.98	27.23	29.47	0.98	28.97	30.32	1.00	29.95	30.63
1.10	29.34	31.85	1.10	31.69	33.26	1.10	32.51	33.25
1.24 a	31.75	. . .	1.20	33.93	35.68	1.18 a	35.03	35.34
1.28	. . .	35.34	1.30	36.14	38.08	1.20	35.03	. . .
			1.38 a	. . .	39.99	1.31 a	37.80	. . .
			1.40 a	38.33	. . .			

	DROP DIAMETER = 5.0MM			DROP DIAMETER = 6.0MM	
θ sec	S, ft		θ sec	S, ft	
	(Flower)	(Liznar)		(Flower)	(Liznar)
0.01			0.01		
0.02	0.778	0.787	0.02	0.644	0.662
0.04	1.533	1.550	0.04	1.280	1.316
0.07	2.628	2.656	0.07	2.218	2.282
0.10	3.680	3.718	0.10	3.144	3.226
0.15	5.352	5.412	0.15	4.638	4.766
0.20	6.94	7.03	0.20	6.136	6.280
0.25	8.48	8.59	0.28	8.44	8.63
0.30	9.97	10.12	0.38	11.25	11.49
0.36	11.71	11.90	0.48	13.99	14.28
0.42	13.41	13.64	0.58	16.69	17.03
0.50	15.65	15.93	0.70 a	19.87	20.28
0.60	18.39	18.74	0.77 a	. . .	22.16
0.70	21.10	21.52	0.80	22.49	. . .
0.80	23.79	24.27	0.91 a	25.45	. . .
0.90	26.47	27.01			
1.00	29.13	29.72			
1.11 a	. . .	32.69			
1.10	31.79	. . .			
1.22 a	34.97	. . .			

a Steady-state attained.

$Nu = 2 R_0 f_c / k_a$. The unit thermal conductance for radiation (f_r) is chosen as 1.05 Btu per (hour) (square foot) (degree Fahrenheit) and is treated as a constant for the system under consideration. The area under the plot ψ vs. θ yields the integral $\int \psi d\theta$, where

$$\psi = \frac{3}{2} (Nu) \left(\frac{\bar{a}}{R_0^2} \right) \left[1 + \frac{1.05}{(Nu)k_a/2R_0} \right] \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad (9)$$

The magnitudes of these integrals are shown in Fig. 17 as a function of time (θ) for various drop diameters, the drops falling through still air from rest, falling in still air from the limiting to the steady-state velocities, and rising in still air from the limiting to zero velocities. The integrals are tabulated in Tables 9, 10 and 11. The data of Liznar were used to establish the kinetic behavior of the drop. Steady-state magnitudes of $\int \psi d\theta$ are tabulated in Table 12, and may be obtained by direct integration (Nu and f_c are constant). Values of ψ for zero velocity were estimated from data by King,[11] employing
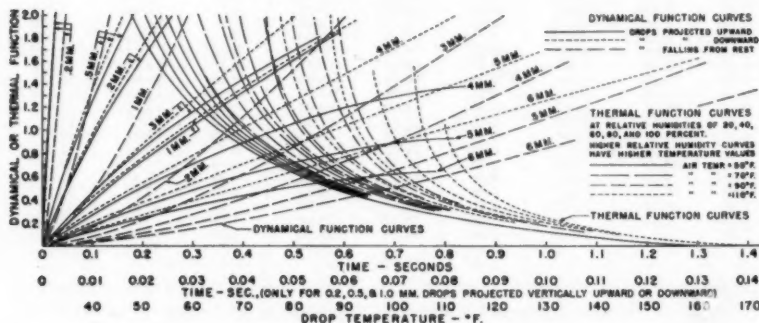
FIG. 17.   EVAPORATIVE COOLING OF DROPS

Dynamical function $\int \psi d\theta$
Thermal function $\int \phi d t$
Drops falling from rest in still air
Drops falling in still air from the limiting velocity to the steady-state velocity
Drops projected vertically upward in still air at the limiting velocity and decelerating to zero velocity

Drop diameters: 0.2, 0.5, 1.0, 2.0, 3.0, 4.0, 5.0 and 6.0 mm
Dry-bulb temperatures: 50, 70, 90, 110 F
Relative humidities: 20, 40, 60, 80, 100 per cent
Initial drop temperature = 170 F

an average magnitude of the free convection modulus times the fluid property modulus (Gr.Pr) for each drop. All properties in these moduli are evaluated for the air-water vapor mixture at conditions corresponding to the arithmetic mean of the temperature of the drop (gaseous side of the liquid-gas interface) and the air-water vapor mixture far away and at the corresponding mean humidities.

Thermal Function

The evaluation of the integral of the right side of Equation (7) for the range of dry-bulb temperatures and relative humidities usually encountered in cooling towers subject to the definition of the ideal system may be also accomplished graphically. The ratio of air-water vapor thermal conductivity to water thermal conductivity is relatively invariable in the temperature range under consideration and has been set equal to 0.420. The Keenan and Keyes Steam Tables (1938) were used as a source for vapor pressure and heat of

[11] King, W. J., *Mechanical Engineering*, 54, 347 (1932).

vaporization data. Results of psychrometric experiments establish (f_c/U') as 0.01635 Btu per (cubic foot) (degree Fahrenheit). Except for the factor $(1 + f_r/f_c)$, the right side of Equation (7), namely,

$$\frac{1}{\left(\frac{k_a}{k_1}\right)\left[(t - t_a) + \dfrac{r(P_w - P_a)}{R_v T_m (f_c/U')(1 + f_r/f_c)}\right]}\, dt = \Phi dt \quad . \quad . \quad . \quad . \quad (10)$$

TABLE 6—VELOCITY-TIME DATA FOR WATER DROPS PROJECTED VERTICALLY UPWARD THROUGH STILL AIR. (DATA OF LIZNAR ONLY.) DECELERATION FROM THE LIMITING TO ZERO VELOCITY

DROP DIAMETER = 0.2MM		DROP DIAMETER = 0.5MM		DROP DIAMETER = 1.0MM		DROP DIAMETER = 2.0MM	
v ft/sec	θ sec	v ft/sec	θ sec	v ft/sec	θ sec	v ft/sec	θ sec
735.0	0	295.0	0	152.0	0	81.0	0
650.0	0.000135	250.0	0.00138	120.0	0.00780	70.0	0.01474
600.0	0.000260	215.0	0.002955	90.0	0.02230	60.0	0.0351
550.0	0.00042	180.0	0.00532	70.0	0.03991	50.0	0.0662
500.0	0.000609	150.0	0.00860	60.0	0.0534	42.0	0.1019
450.0	0.00085	110.0	0.01590	50.0	0.0721	36.0	0.1385
400.0	0.001165	75.0	0.02805	40.0	0.0987	28.8	0.2070
350.0	0.00155	50.0	0.04604	32.0	0.1297	20.0	0.3052
300.0	0.00208	40.0	0.0594	24.0	0.1766	15.0	0.3901
260.0	0.00266	30.0	0.0804	16.0	0.2535	12.0	0.4527
220.0	0.00353	22.0	0.1070	12.0	0.3105	10.0	0.4991
180.0	0.00473	15.0	0.1453	10.0	0.3456	8.0	0.5495
140.0	0.00651	10.0	0.1908	8.0	0.3857	6.0	0.6038
100.0	0.00973	8.0	0.2162	6.0	0.4312	4.0	0.6612
75.0	0.01317	6.0	0.2475	4.0	0.4825	2.0	0.7212
65.0	0.01531	4.0	0.2865	2.0	0.5391	0.0	0.7829
40.0	0.02423	2.0	0.3352	0.0	0.6000		
25.0	0.03574	1.0	0.3631				
18.0	0.0458	0.0	0.3933				
12.0	0.0603						
8.0	0.0763						
6.0	0.0883						
4.0	0.1056						
2.0	0.1337						
1.0	0.1536						
0.0	0.1807						

DROP DIAMETER = 3.0MM		DROP DIAMETER = 4.0MM		DROP DIAMETER = 5.0MM		DROP DIAMETER = 6.0MM	
v ft/sec	θ sec	v ft/sec	θ sec	v ft/sec	θ sec	v ft/sec	θ sec
58.9	0	47.0	0	40.0	0	33.4	0
55.0	0.01287	44.0	0.01574	37.0	0.0240	31.0	0.02855
50.0	0.03180	40.0	0.0410	34.0	0.0517	28.0	0.0694
42.0	0.0739	35.0	0.0832	30.0	0.0979	24.0	0.1352
36.0	0.1187	30.0	0.1392	26.0	0.1566	20.0	0.2152
28.0	0.2038	26.0	0.1957	22.0	0.2283	16.0	0.3089
24.0	0.2601	22.0	0.2639	18.0	0.3130	12.0	0.4145
20.0	0.3275	18.0	0.3446	14.0	0.4102	10.0	0.4708
16.0	0.4067	14.0	0.4376	10.0	0.5178	8.0	0.5289
12.0	0.4989	10.0	0.5419	8.0	0.5749	6.0	0.5885
10.0	0.5498	8.0	0.5980	6.0	0.6340	4.0	0.6492
8.0	0.6039	6.0	0.6562	4.0	0.6944	2.0	0.7108
6.0	0.6607	4.0	0.7162	2.0	0.7559	0.0	0.7728
4.0	0.7199	2.0	0.7774	0.0	0.8179		
2.0	0.7808	0.0	0.8394				
0.0	0.8426						

TABLE 7—DISTANCE-TIME DATA FOR WATER DROPS PROJECTED VERTICALLY UPWARD
THROUGH STILL AIR. (DATA OF LIZNAR ONLY)

DROP DIAMETER = 0.2MM		DROP DIAMETER = 0.5MM		DROP DIAMETER = 1.0MM		DROP DIAMETER = 2.0MM	
θ sec	S ft	θ sec	S ft	θ sec	S ft	θ sec	S ft
0.001	0.5460	0.001	0.2768	0.005	0.7015	0.01	0.773
0.002	0.9052	0.002	0.5256	0.01	1.3119	0.02	1.475
0.003	1.1744	0.003	0.7504	0.02	2.346	0.04	2.719
0.004	1.3968	0.0044	1.0352	0.03	3.211	0.06	3.809
0.005	1.584	0.0064	1.396	0.04	3.961	0.08	4.789
0.0064	1.802	0.0086	1.745	0.05	4.622	0.10	5.678
0.008	2.011	0.015	2.581	0.06	5.216	0.12	6.487
0.00973	2.198	0.025	3.542	0.08	6.239	0.14	7.23
0.015	2.621	0.035	4.258	0.10	7.08	0.16	7.92
0.025	3.133	0.050	5.067	0.13	8.16	0.18	8.56
0.035	3.450	0.065	5.689	0.16	9.04	0.20	9.15
0.050	3.756	0.080	6.191	0.20	9.98	0.24	10.22
0.065	3.951	0.100	6.734	0.25	10.91	0.28	11.16
0.080	4.083	0.13	7.35	0.30	11.63	0.32	11.98
0.100	4.201	0.16	7.81	0.35	12.19	0.36	12.69
0.125	4.288	0.20	8.25	0.40	12.62	0.40	13.31
0.150	4.334	0.25	8.62	0.45	12.93	0.44	13.85
0.1807	4.354 [*]	0.30	8.85	0.50	13.14	0.48	14.32
		0.35	8.97	0.55	13.27	0.54	14.89
		0.3933	9.01 [*]	0.60	13.31 [*]	0.60	15.33
						0.66	15.63
						0.72	15.81
						0.7829	15.88 [*]

DROP DIAMETER = 3.0MM		DROP DIAMETER = 4.0MM		DROP DIAMETER = 5.0MM		DROP DIAMETER = 6.0MM	
θ sec	S ft	θ sec	S ft	θ sec	S ft	θ sec	S ft
0.01	0.58	0.01	0.46	0.01	0.39	0.01	0.32
0.02	1.109	0.02	0.901	0.02	0.775	0.02	0.652
0.04	2.119	0.04	1.734	0.04	1.503	0.04	1.270
0.06	3.043	0.06	2.510	0.06	2.186	0.06	1.858
0.08	3.897	0.08	3.238	0.08	2.832	0.08	2.417
0.10	4.691	0.10	3.925	0.10	3.444	0.10	2.950
0.12	5.432	0.12	4.605	0.12	4.025	0.12	3.459
0.14	6.128	0.14	5.280	0.14	4.579	0.14	3.945
0.16	6.783	0.16	5.874	0.16	5.107	0.16	4.409
0.18	7.40	0.18	6.428	0.18	5.611	0.18	4.853
0.20	7.98	0.20	6.96	0.20	6.132	0.20	5.276
0.24	9.05	0.24	7.97	0.24	7.03	0.24	6.066
0.28	10.02	0.28	8.86	0.28	7.85	0.28	6.79
0.32	10.88	0.32	9.66	0.32	8.59	0.32	7.44
0.36	11.65	0.36	10.39	0.38	9.58	0.38	8.31
0.40	12.34	0.40	11.04	0.44	10.42	0.44	9.04
0.46	13.24	0.46	11.90	0.50	11.13	0.50	9.64
0.52	13.98	0.52	12.62	0.58	11.87	0.58	10.25
0.58	14.58	0.58	13.21	0.66	12.39	0.68	10.71
0.66	15.18	0.66	13.79	0.74	12.70	0.7728	10.85 [*]
0.74	15.55	0.74	14.15	0.8179	12.80 [*]		
0.8426	15.72 [*]	0.8394	14.32 [*]				

[*] Maximum distance of rise.

depends only on the thermal conditions of the drop for a given partial pressure
of the water vapor far away. In order to separate the dynamical and thermal
effects, the factor $(1 + f_r/f_c)$ is here set equal to 1.03 as a working approxi-
mation. An initial drop temperature of 170 F was chosen as representative of
the maximum temperature at which water is delivered to the system, but the

curves are applicable to any initial temperature below this magnitude. The integral $\int_{170}^{t} \Phi dt$ is revealed graphically in Fig. 17, and presented in Table 13. For points beyond the range of the plot, direct interpolation of data in Tables 9, 10, 11 and 13 will yield satisfactory accuracy.

Application of the ψ and Φ Functions to the Solution of the Drop Cooling Problem

1. *A generalized comparative representation of the temperature-time relations:* The temperature-time history of a water drop under any combination of dynamical and thermal conditions within the range studied may be obtained from Fig. 17. The dynamical and thermal functions, $\int \psi d\theta$ and $\int \Phi dt$ respectively, are plotted as ordinates to the same scale. Elapsed time serves as the abscissa for the dynamical curves, and drop temperature serves likewise for the thermal curves. In accord with the solution indicated by Equation (7), the drop temperature is found as a function of elapsed time for a given vertical trajectory and drop size in the following manner: For a given interval of time, the increment in the dynamical function is indicated directly by the curve corresponding to the drop diameter and trajectory under consideration. Then, for a given initial drop temperature and air state, the final drop temperature corresponding to the given change in the dynamical function is obtained by increasing the ordinate (the initial point on the thermal function curve is fixed by the initial drop temperature) of the particular thermal function curve which corresponds to the air state far away by the amount of the specified dynamical function increment and reading the drop temperature as the abscissa of the point so determined.

The lower cooling limit of the thermal function curves is that drop temperature for which the quantity ψ is infinite, i.e., the temperature which prevails independent of time when a balance has been established between the rates of heat loss and gain by the drop. Actual equipments are seldom, if ever, designed to attain this temperature because of the relative ineffectiveness of the cooling process in the region of close approach and the consequent large size of transfer chamber required. The lower limiting temperature is closely defined by the wet-bulb temperature of the air-vapor mixture involved, magnitudes of which as obtained from a psychrometric chart are given in the following tabulation:

WET-BULB TEMPERATURES

DRY-BULB, F	R. H. PER CENT				
	20	40	60	80	100
50	36.8	40.5	43.9	46.9	50
70	50.1	55.9	61.0	65.8	70
90	66.0	71.2	78.3	84.5	90
110	75.4	86.6	95.6	103.5	110

TABLE 9—DYNAMICAL FUNCTION FOR THE EVAPORATIVE COOLING OF WATER DROPS FALLING FROM REST THROUGH STILL AIR. (DATA OF LIZNAR)

DROP DIAMETER = 0.2MM		DROP DIAMETER = 0.5MM		DROP DIAMETER = 1.0MM		DROP DIAMETER = 2.0MM	
θ sec	$\int \psi d\theta$	θ sec	$\int \psi d\theta$	θ sec	$\int \psi d\theta$	θ sec	$\int \psi d\theta$
0.01	0.268	0.01	0.0593	0.01	0.0219	0.02	0.0155
0.02	0.669	0.02	0.1498	0.02	0.0534	0.04	0.0420
0.04	1.663	0.04	0.386	0.04	0.1362	0.08	0.1142
0.06	2.820	0.06	0.674	0.06	0.234	0.12	0.204
0.08	4.083	0.08	1.000	0.08	0.345	0.16	0.306
0.10	5.409	0.10	1.357	0.10	0.468	0.20	0.420
0.12	6.775	0.14	2.139	0.14	0.742	0.28	0.677
0.14	8.171	0.18	2.991	0.20	1.210	0.36	0.965
0.16	9.589	0.22	3.897	0.26	1.731	0.44	1.280
0.18	11.026	0.26	4.845	0.32	2.296	0.52	1.616
0.20	12.514	0.30	5.825	0.38	2.897	0.60	1.970
0.24 ᵃ	15.455	0.34	6.827	0.44	3.524	0.68	2.337
		0.38	7.840	0.50	4.171	0.76	2.714
		0.42	8.873	0.56	4.833	0.84	3.101
		0.46	9.912	0.64	5.732	0.94	3.593
		0.50	10.958	0.72	6.646	1.04	4.094
		0.56	12.538	0.80	7.574	1.16	4.704
		0.62	14.130	0.90	8.750	1.28	5.321
		0.68 ᵃ	15.729	1.01 ᵃ	10.062	1.40	5.942
						1.52	6.566
						1.66 ᵃ	7.298

DROP DIAMETER = 3.0MM		DROP DIAMETER = 4.0MM		DROP DIAMETER = 5.0MM		DROP DIAMETER = 6.0MM	
θ sec	$\int \psi d\theta$	θ sec	$\int \psi d\theta$	θ sec	$\int \psi d\theta$	θ sec	$\int \psi d\theta$
0.02	0.0087	0.02	0.0060	0.02	0.0040	0.02	0.0037
0.04	0.0230	0.04	0.0154	0.04	0.0105	0.04	0.0088
0.08	0.0617	0.08	0.0406	0.08	0.0285	0.08	0.0225
0.12	0.1103	0.12	0.0723	0.12	0.0512	0.12	0.0398
0.16	0.166	0.16	0.1090	0.16	0.0774	0.16	0.0600
0.20	0.229	0.20	0.150	0.20	0.1067	0.20	0.0827
0.28	0.372	0.28	0.244	0.28	0.175	0.28	0.1348
0.36	0.535	0.36	0.351	0.36	0.253	0.36	0.195
0.44	0.716	0.44	0.470	0.44	0.340	0.44	0.261
0.52	0.910	0.52	0.598	0.52	0.433	0.52	0.333
0.60	1.115	0.60	0.734	0.60	0.533	0.60	0.410
0.68	1.330	0.68	0.877	0.68	0.638	0.68	0.492
0.76	1.551	0.76	1.027	0.76	0.748	0.76	0.577
0.86	1.837	0.86	1.221	0.86	0.890	0.86	0.687
0.96	2.130	0.96	1.422	0.96	1.037	0.96	0.801
1.06	2.430	1.06	1.625	1.06	1.188	1.06	0.918
1.16	2.797	1.16	1.834	1.16	1.341	1.18	1.061
1.28	3.170	1.28	2.092	1.28	1.529	1.28	1.182
1.40	3.546	1.40	2.343	1.40	1.718	1.40	1.327
1.58	4.116	1.58	2.732	1.60	2.037	1.60	1.572
1.76	4.691	1.76	3.126	1.80	2.161	1.80	1.819
1.90 ᵃ	5.141	1.92 ᵃ	3.479	1.85 ᵃ	2.443	2.00 ᵃ	2.066

ᵃ Steady-state velocity attained.

smaller than the limiting velocity. The previously determined velocity-distance-time data and curves present the necessary information for the execution of such calculations.

For the case in which drops are descending or ascending through air the temperature and humidity of which are a space function, the ordinate increment obtained from the dynamical function is added to the initial ordinate (located on the curve defined by the initial air temperature and relative humidity

TABLE 10—DYNAMICAL FUNCTION FOR THE EVAPORATIVE COOLING OF WATER DROPS PROJECTED VERTICALLY DOWNARD THROUGH STILL AIR. (DATA OF LIZNAR)

DROP DIAMETER = 0.2MM		DROP DIAMETER = 0.5MM		DROP DIAMETER = 1.0MM		DROP DIAMETER = 2.0MM	
θ sec	$\int \psi d\theta$	θ sec	$\int \psi d\theta$	θ sec	$\int \psi d\theta$	θ sec	$\int \psi d\theta$
0.0002	0.234	0.001	0.161	0.01	0.407	0.0125	0.13
0.0004	0.417	0.002	0.312	0.02	0.759	0.0250	0.26
0.0008	0.757	0.004	0.596	0.03	1.075	0.0375	0.388
0.0012	1.067	0.006	0.856	0.04	1.369	0.0500	0.508
0.0016	1.355	0.008	1.099	0.06	1.910	0.0625	0.625
0.0020	1.625	0.010	1.329	0.08	2.400	0.0750	0.735
0.0028	2.123	0.02	2.347	0.10	2.851	0.1000	0.953
0.0036	2.576	0.03	3.175	0.12	3.269	0.125	1.159
0.0044	2.996	0.04	3.845	0.14	3.661	0.150	1.355
0.0052	3.388	0.05	4.427	0.16	4.050	0.175	1.546
0.0060	3.755	0.06	4.969	0.18	4.400	0.200	1.730
0.0070	4.187	0.08	5.969	0.20	4.734	0.25	2.081
0.0080	4.594	0.10	6.893	0.24	5.368	0.30	2.416
0.0100	5.328	0.14	8.581	0.28	5.966	0.35	2.739
0.012	5.487	0.18	10.097	0.32	6.540	0.40	3.054
0.020	7.811	0.22	10.789	0.38	7.373	0.45	3.355
0.030	10.235	0.26	11.433	0.44	8.183	0.50	3.659
0.040	12.199	0.30	12.045	0.52	9.232	0.55	3.953
0.050	13.779	0.34	13.221	0.60	10.054	0.60	4.241
0.060	15.159	0.38	13.795	0.70	11.498	0.65	4.523
0.080	17.591	0.42	14.359	0.80	12.718	0.70	4.806
0.100	19.756	0.50	16.570	0.90	13.933	0.80	5.359
0.12	22.123	0.58	18.723	0.96 ª	14.642	0.90	5.903
0.14	23.930	0.66 ª	20.865			1.00	6.441
0.16	25.620					1.10	6.973
0.18	27.238					1.20	7.500
0.20	28.807					1.28 ª	7.919
0.24	31.868						
0.28	34.871						
0.34 ª	39.303						

DROP DIAMETER = 3.0MM		DROP DIAMETER = 4.0MM		DROP DIAMETER = 5.0MM		DROP DIAMETER = 6.0MM	
θ sec	$\int \psi d\theta$	θ sec	$\int \psi d\theta$	θ sec	$\int \psi d\theta$	θ sec	$\int \psi d\theta$
0.0125	0.0629	0.025	0.0734	0.05	0.0969	0.05	0.0683
0.0250	0.1238	0.050	0.1448	0.10	0.1905	0.10	0.1353
0.050	0.241	0.075	0.214	0.15	0.281	0.20	0.267
0.075	0.355	0.100	0.282	0.20	0.370	0.30	0.396
0.100	0.464	0.125	0.348	0.25	0.457	0.40	0.523
0.125	0.570	0.150	0.413	0.30	0.543	0.50	0.649
0.150	0.674	0.175	0.478	0.35	0.628	0.60	0.774
0.175	0.775	0.200	0.540	0.40	0.715	0.77 ª	0.985
0.200	0.874	0.25	0.667	0.50	0.879	0.97	1.2225
0.25	1.068	0.30	0.789	0.60	1.044		
0.30	1.257	0.35	0.909	0.70	1.208		
0.35	1.442	0.40	1.028	0.80	1.372		
0.40	1.623	0.45	1.145	0.90	1.535		
0.45	1.800	0.50	1.262	1.00	1.697		
0.50	1.975	0.55	1.377	1.11 ª	1.875		
0.55	2.149	0.60	1.491				
0.60	2.320	0.70	1.718				
0.70	2.660	0.80	1.943				
0.80	2.994	0.90	2.165				
0.90	3.326	1.00	2.387				
1.00	3.655	1.10	2.608				
1.10	3.981	1.18 ª	2.825				
1.20	4.306						
1.38 ª	4.888						

ª Steady-state velocity attained.

TABLE 9—DYNAMICAL FUNCTION FOR THE EVAPORATIVE COOLING OF WATER DROPS FALLING FROM REST THROUGH STILL AIR. (DATA OF LIZNAR)

DROP DIAMETER = 0.2MM		DROP DIAMETER = 0.5MM		DROP DIAMETER = 1.0MM		DROP DIAMETER = 2.0MM	
θ sec	$\int \psi d\theta$	θ sec	$\int \psi d\theta$	θ sec	$\int \psi d\theta$	θ sec	$\int \psi d\theta$
0.01	0.268	0.01	0.0593	0.01	0.0219	0.02	0.0155
0.02	0.669	0.02	0.1498	0.02	0.0534	0.04	0.0420
0.04	1.663	0.04	0.386	0.04	0.1362	0.08	0.1142
0.06	2.820	0.06	0.674	0.06	0.234	0.12	0.204
0.08	4.083	0.08	1.000	0.08	0.345	0.16	0.306
0.10	5.409	0.10	1.357	0.10	0.468	0.20	0.420
0.12	6.775	0.14	2.139	0.14	0.742	0.28	0.677
0.14	8.171	0.18	2.991	0.20	1.210	0.36	0.965
0.16	9.589	0.22	3.897	0.26	1.731	0.44	1.280
0.18	11.026	0.26	4.845	0.32	2.296	0.52	1.616
0.20	12.514	0.30	5.825	0.38	2.897	0.60	1.970
0.24ᵃ	15.455	0.34	6.827	0.44	3.524	0.68	2.337
		0.38	7.840	0.50	4.171	0.76	2.714
		0.42	8.873	0.56	4.833	0.84	3.101
		0.46	9.912	0.64	5.732	0.94	3.593
		0.50	10.958	0.72	6.646	1.04	4.094
		0.56	12.538	0.80	7.574	1.16	4.704
		0.62	14.130	0.90	8.750	1.28	5.321
		0.68ᵃ	15.729	1.01ᵃ	10.062	1.40	5.942
						1.52	6.566
						1.66ᵃ	7.298

DROP DIAMETER = 3.0MM		DROP DIAMETER = 4.0MM		DROP DIAMETER = 5.0MM		DROP DIAMETER = 6.0MM	
θ sec	$\int \psi d\theta$	θ sec	$\int \psi d\theta$	θ sec	$\int \psi d\theta$	θ sec	$\int \psi d\theta$
0.02	0.0087	0.02	0.0060	0.02	0.0040	0.02	0.0037
0.04	0.0230	0.04	0.0154	0.04	0.0105	0.04	0.0088
0.08	0.0617	0.08	0.0406	0.08	0.0285	0.08	0.0225
0.12	0.1103	0.12	0.0723	0.12	0.0512	0.12	0.0398
0.16	0.166	0.16	0.1090	0.16	0.0774	0.16	0.0600
0.20	0.229	0.20	0.150	0.20	0.1067	0.20	0.0827
0.28	0.372	0.28	0.244	0.28	0.175	0.28	0.1348
0.36	0.535	0.36	0.351	0.36	0.253	0.36	0.195
0.44	0.716	0.44	0.470	0.44	0.340	0.44	0.261
0.52	0.910	0.52	0.598	0.52	0.433	0.52	0.333
0.60	1.115	0.60	0.734	0.60	0.533	0.60	0.410
0.68	1.330	0.68	0.877	0.68	0.638	0.68	0.492
0.76	1.551	0.76	1.027	0.76	0.748	0.76	0.577
0.86	1.837	0.86	1.221	0.86	0.890	0.86	0.687
0.96	2.130	0.96	1.422	0.96	1.037	0.96	0.801
1.06	2.430	1.06	1.625	1.06	1.188	1.06	0.918
1.16	2.797	1.16	1.834	1.16	1.341	1.18	1.061
1.28	3.170	1.28	2.092	1.28	1.529	1.28	1.182
1.40	3.546	1.40	2.343	1.40	1.718	1.40	1.327
1.58	4.116	1.58	2.732	1.60	2.037	1.60	1.572
1.76	4.691	1.76	3.126	1.80	2.161	1.80	1.819
1.90ᵃ	5.141	1.92ᵃ	3.479	1.85ᵃ	2.443	2.00ᵃ	2.066

ᵃ Steady-state velocity attained.

smaller than the limiting velocity. The previously determined velocity-distance-time data and curves present the necessary information for the execution of such calculations.

For the case in which drops are descending or ascending through air the temperature and humidity of which are a space function, the ordinate increment obtained from the dynamical function is added to the initial ordinate (located on the curve defined by the initial air temperature and relative humidity

TABLE 10—DYNAMICAL FUNCTION FOR THE EVAPORATIVE COOLING OF WATER DROPS PROJECTED VERTICALLY DOWNARD THROUGH STILL AIR. (DATA OF LIZNAR)

DROP DIAMETER = 0.2MM		DROP DIAMETER = 0.5MM		DROP DIAMETER = 1.0MM		DROP DIAMETER = 2.0MM	
θ sec	$\int \psi d\theta$	θ sec	$\int \psi d\theta$	θ sec	$\int \psi d\theta$	θ sec	$\int \psi d\theta$
0.0002	0.234	0.001	0.161	0.01	0.407	0.0125	0.135
0.0004	0.417	0.002	0.312	0.02	0.759	0.0250	0.264
0.0008	0.757	0.004	0.596	0.03	1.075	0.0375	0.388
0.0012	1.067	0.006	0.856	0.04	1.369	0.0500	0.508
0.0016	1.355	0.008	1.099	0.06	1.910	0.0625	0.625
0.0020	1.625	0.010	1.329	0.08	2.400	0.0750	0.735
0.0028	2.123	0.02	2.347	0.10	2.851	0.1000	0.953
0.0036	2.576	0.03	3.175	0.12	3.269	0.125	1.159
0.0044	2.996	0.04	3.845	0.14	3.661	0.150	1.355
0.0052	3.388	0.05	4.427	0.16	4.050	0.175	1.546
0.0060	3.755	0.06	4.969	0.18	4.400	0.200	1.730
0.0070	4.187	0.08	5.969	0.20	4.734	0.25	2.081
0.0080	4.594	0.10	6.893	0.24	5.368	0.30	2.416
0.0100	5.328	0.14	8.581	0.28	5.966	0.35	2.739
0.012	5.487	0.18	10.097	0.32	6.540	0.40	3.054
0.020	7.811	0.22	10.789	0.38	7.373	0.45	3.355
0.030	10.235	0.26	11.433	0.44	8.183	0.50	3.659
0.040	12.199	0.30	12.045	0.52	9.232	0.55	3.953
0.050	13.779	0.34	13.221	0.60	10.054	0.60	4.241
0.060	15.159	0.38	13.795	0.70	11.498	0.65	4.523
0.080	17.591	0.42	14.359	0.80	12.718	0.70	4.806
0.100	19.756	0.50	16.570	0.90	13.933	0.80	5.359
0.12	22.123	0.58	18.723	0.96 [a]	14.642	0.90	5.903
0.14	23.930	0.66 [a]	20.865			1.00	6.441
0.16	25.620					1.10	6.973
0.18	27.238					1.20	7.500
0.20	28.807					1.28 [a]	7.919
0.24	31.868						
0.28	34.871						
0.34 [a]	39.303						

DROP DIAMETER = 3.0MM		DROP DIAMETER = 4.0MM		DROP DIAMETER = 5.0MM		DROP DIAMETER = 6.0MM	
θ sec	$\int \psi d\theta$	θ sec	$\int \psi d\theta$	θ sec	$\int \psi d\theta$	θ sec	$\int \psi d\theta$
0.0125	0.0629	0.025	0.0734	0.05	0.0969	0.05	0.0683
0.0250	0.1238	0.050	0.1448	0.10	0.1905	0.10	0.1353
0.050	0.241	0.075	0.214	0.15	0.281	0.20	0.267
0.075	0.355	0.100	0.282	0.20	0.370	0.30	0.396
0.100	0.464	0.125	0.348	0.25	0.457	0.40	0.523
0.125	0.570	0.150	0.413	0.30	0.543	0.50	0.649
0.150	0.674	0.175	0.478	0.35	0.628	0.60	0.774
0.175	0.775	0.200	0.540	0.40	0.715	0.77 [a]	0.985
0.200	0.874	0.25	0.667	0.50	0.879		
0.25	1.068	0.30	0.789	0.60	1.044	0.97	1.2225
0.30	1.257	0.35	0.909	0.70	1.208		
0.35	1.442	0.40	1.028	0.80	1.372		
0.40	1.623	0.45	1.145	0.90	1.535		
0.45	1.800	0.50	1.262	1.00	1.697		
0.50	1.975	0.55	1.377	1.11 [a]	1.875		
0.55	2.149	0.60	1.491				
0.60	2.320	0.70	1.718				
0.70	2.660	0.80	1.943				
0.80	2.994	0.90	2.165				
0.90	3.326	1.00	2.387				
1.00	3.655	1.10	2.608				
1.10	3.981	1.18 [a]	2.825				
1.20	4.306						
1.38 [a]	4.888						

[a] Steady-state velocity attained.

Table 11—Dynamical Function for the Evaporative Cooling of Water Drops Projected Vertically Upward Through Still Air. (Data of Liznar)

Drop Diameter = 0.2mm		Drop Diameter = 0.5mm		Drop Diameter = 1.0mm		Drop Diameter = 2.0mm	
θ sec	$\int \psi d\theta$	θ sec	$\int \psi d\theta$	θ sec	$\int \psi d\theta$	θ sec	$\int \psi d\theta$
0.0004	0.382	0.0008	0.130	0.01	0.386	0.01	0.1068
0.0012	1.031	0.0020	0.311	0.02	0.732	0.02	0.2085
0.0020	1.591	0.0044	0.633	0.04	1.338	0.04	0.3992
0.0028	2.089	0.0068	0.913	0.06	1.872	0.06	0.5779
0.0036	2.544	0.0100	1.244	0.08	2.352	0.08	0.7473
0.0048	3.165	0.015	1.734	0.10	2.792	0.10	0.9086
0.0064	3.906	0.020	2.178	0.12	3.198	0.12	1.0625
0.0080	4.512	0.030	2.982	0.14	3.574	0.14	1.2094
0.0104	5.470	0.040	3.704	0.16	3.926	0.16	1.3505
0.0128	6.279	0.050	4.362	0.18	4.256	0.20	1.6176
0.0150	6.964	0.060	4.969	0.20	4.566	0.24	1.8663
0.020	8.381	0.070	5.535	0.24	5.142	0.28	2.0978
0.025	9.621	0.080	6.066	0.28	5.662	0.34	2.4164
0.030	10.741	0.100	7.040	0.32	6.136	0.40	2.7031
0.040	12.651	0.120	7.916	0.38	6.770	0.46	2.9600
0.050	14.339	0.140	8.712	0.44	7.316	0.52	3.1884
0.060	15.930	0.160	9.841	0.50	7.768	0.58	3.3892
0.070	17.160	0.180	10.515	0.54	8.010	0.64	3.5633
0.080	18.360	0.210	11.438	0.58	8.190	0.68	3.6622
0.090	19.453	0.250	12.520	0.60 [a]	8.244	0.72	3.7452
0.100	20.451	0.300	13.636			0.74	3.7798
0.120	22.190	0.350	14.495			0.76	3.8106
0.140	23.622	0.360	14.633			0.7829 [a]	3.8328
0.150	24.191	0.370	14.758				
0.160	24.708	0.380	14.864				
0.1807 [a]	25.374	0.3933 [a]	14.955				

Drop Diameter = 3mm		Drop Diameter = 4mm		Drop Diameter = 5mm		Drop Diameter = 6mm	
θ sec	$\int \psi d\theta$	θ sec	$\int \psi d\theta$	θ sec	$\int \psi d\theta$	θ sec	$\int \psi d\theta$
0.02	0.0999	0.02	0.0585	0.02	0.0389	0.02	0.0272
0.04	0.1946	0.04	0.1145	0.04	0.0765	0.04	0.0536
0.06	0.2850	0.06	0.1685	0.06	0.1130	0.08	0.1047
0.08	0.3718	0.08	0.2209	0.10	0.1830	0.12	0.1534
0.10	0.4556	0.10	0.2718	0.14	0.2494	0.18	0.2224
0.14	0.6149	0.14	0.3694	0.20	0.3433	0.24	0.2881
0.18	0.7649	0.18	0.4620	0.28	0.4588	0.30	0.3479
0.22	0.9062	0.22	0.5500	0.36	0.5633	0.38	0.4211
0.26	1.0395	0.28	0.6739	0.44	0.6574	0.44	0.4709
0.32	1.2259	0.34	0.7887	0.52	0.7411	0.52	0.5301
0.38	1.3969	0.40	0.8949	0.58	0.7967	0.58	0.5687
0.46	1.6023	0.48	1.0237	0.64	0.8453	0.62	0.5915
0.54	1.7833	0.56	1.1372	0.70	0.8863	0.66	0.6115
0.60	1.9029	0.62	1.2113	0.74	0.9088	0.70	0.6284
0.66	2.0075	0.68	1.2755	0.78	0.9266	0.72	0.6355
0.72	2.0963	0.74	1.3287	0.80	0.9332	0.74	0.6415
0.76	2.1256	0.78	1.3571	0.8179 [a]	0.9371	0.76	0.6463
0.80	2.1857	0.80	1.3687			0.7728 [a]	0.6484
0.82	2.2012	0.82	1.3783				
0.8426 [a]	2.2130	0.8394 [a]	1.3846				

[a] Maximum distance of rise attained.

parameter) corresponding to the initial drop temperature. The intersection of the ordinate increment with the thermal function curve defined by the air temperature and relative humidity at the new position of the drop in the system (i.e., $S + \Delta S$ corresponding to $\theta + \Delta\theta$) yields the drop temperature in the new environment.
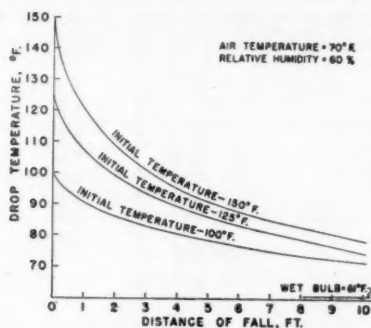
FIG. 18. EVAPORATIVE COOLING OF A
5 MM WATER DROP FALLING FROM REST
THROUGH STILL AIR
(AIR TEMPERATURE = 70 F, R. R. = 60
PER CENT, W. B. = 61 F)

The comparative shape and location of the curves in Fig. 17 serve to indicate performance trends.   The very rapid cooling possible with small drops discharged from a nozzle is most striking.   The increasing spread between the curves for the larger drops is to be interpreted for any particular application with due regard for the distance-time curves, for distance is an important design variable.   At the boiling point the thermal function curves become parellel to the temperature axis, indicating an instantaneous change of state.   The slope of the thermal function curves as dependent upon the drop temperature and air
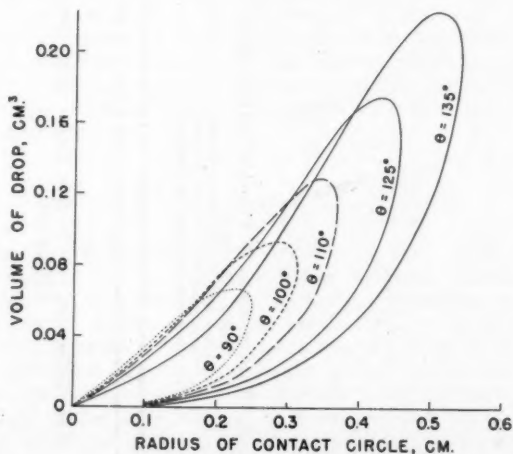
FIG. 19. VOLUME OF PENDANT DROP AS FUNCTION OF
RADIUS OF CONTACT FOR VARIOUS CONTACT ANGLES
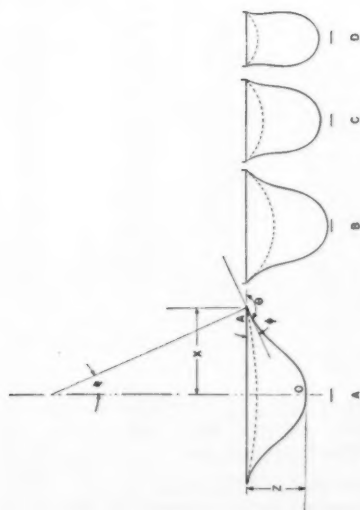
FIG. 20. TYPICAL FORMS OF PENDANT WATER DROPS

A. Infinite glass plane.
B. Glass tube of 0.9525 cm diameter.
C. Glass tube of 0.724 cm diameter.
D. Glass tube of 0.448 cm diameter.
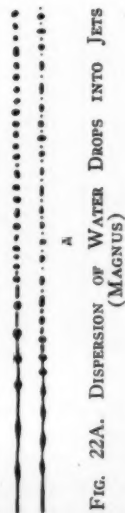
FIG. 22A. DISPERSION OF WATER DROPS INTO JETS (MAGNUS)

FIG. 22B. SHAPE OF A DROP FALLING AT THE STEADY-STATE VELOCITY (FLOWER)

CONTACT ANGLE, DEGREES
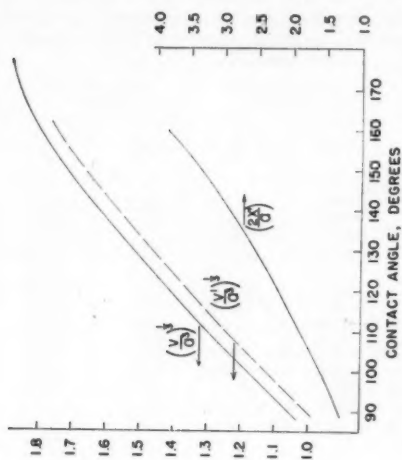
FIG. 21. CRITICAL FUNCTIONS OF PENDANT DROPS

A. Dispersion into drops without the influence of air.

B. Dispersion into drops under the influence of air.

C. Dispersion into waves under the influence of air.
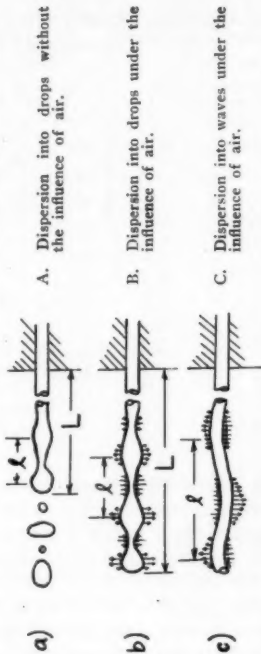
FIG. 23. PHENOMENA OF JET DISPERSION

state is indicative of the cooling rate in terms of operating conditions. A higher initial temperature leads to a more rapid initial cooling for the same dynamical conditions.

2. *Illustrative example:* Consider the case in which water drops of 5 mm diameter form on the underside of the slats of an atmospheric type cooling tower deck. The drop temperature as a function of distance of fall (deck spacing) for initial temperatures of 150, 125, and 100 F is desired. The effective air state in the interdeck space is 70 F and 60 per cent relative humidity. As a limiting design condition still air will be postulated.

The elapsed time corresponding to successive distances of fall is obtained from Fig. 3 (Data of Liznar). Values of the dynamical function corresponding to these time intervals are read on Fig. 17 and are then applied as increments to the ordinate (the initial ordinate corresponds to the initial drop temperature) of the thermal function curve which is presented on the same plot for each of the specified initial temperatures. Successive magnitudes of the drop temperature are thereby determined directly.

Calculations are summarized in Table 11 and the predicted performance is plotted in Fig. 18. An immediate observation is that if the drop starting at 150 F were stopped at the distances of fall corresponding to 125 and 100 F its temperature after starting again would be lower than if it had continued in the uninterrupted trajectory. An interesting and practical application study is here indicated. Knowledge of the temperature changes in various types of drop impingement and re-formation processes is needed, however, to completely specify the behavior.

3. *General comments:* The application of the dynamical and thermal functions to the prediction of drop cooling should be made advisedly. Experimental proof of the validity of the computations must be awaited. Johnstone and Williams [12] present distance-velocity-time curves for drops ejected from a 3/16-in. lava nozzle operating at 35 lb per square inch and discharging into a counter current gas stream moving at 5 fps. These authors also present drop size distribution data and develop a method of calculating the volume mass transfer coefficient which includes the probability of coalescence of the drops as they proceed along their trajectory.

FORMATION OF DROPS FROM LIQUID BODIES

The process of drop formation from a larger body of liquid requires the expenditure of energy for the creation of a new liquid-gas interfacial surface. This energy transfer may take place either through a spontaneous dissipation of mechanical energy on the part of the parent liquid body, or through the action of external agencies in creating conditions of surface instability which result in the separation of particles from the surface, or through a dynamic combination of the two operations. The first case is exemplified by a pressure atomizing nozzle or the shattering of a stream or drop upon impact; the second occurs when a gas stream passes at a high velocity over a liquid surface or when the action of gravity causes a pendant drop to fall from the underside

[12] Johnstone, H. F. and Williams, G. C., *Industrial and Engineering Chemistry*, 31, 993 (1939).

of a wetted surface; the third is the most common action, being found in the dispersion of all types of liquid jets and the operation of film-forming nozzles. Data will be presented for water as the liquid, but the data are applicable to the behavior of other liquids.

Drop formation from the underside of a wetted solid object when the rate of flow is not sufficient to produce a jet.

Quantitative analysis of this problem is developed through a study [13] of the critical equilibrium condition between the weight of liquid suspended in a form which is about to separate from the main body as a drop and the forces of surface tension striving to maintain a single body. The maximum volume of a drop which may form in this manner is determined by the angle between the liquid surface at the plane of separation and the horizontal. The nature and contour of the wetted surface affect the size of the drop produced as they modify this angle. The predicted drop volume represents the maximum because any disturbing influences such as air motion against the forming drop or vibration of the supporting surface, will disrupt the static capillary equilibrium and result in the attainment of an unstable condition with a smaller drop weight than would otherwise obtain.

An analysis of the forces acting upon the pendant drop results in the following equation for the volume of the drop:

$$V = \frac{\pi b^2 x^2}{\beta} \left[\frac{1}{\sigma} - \frac{Sin\phi}{x} \right]$$

where $\quad -\beta = \dfrac{b^2(\delta_1 - \delta_g)}{H_{gl}}$ . . . . (11)

The computations for the maximum (critical) drop volume (V_{max}) as a function of the contact angle (θ, Fig. 20) have been accomplished by Fritz [14] and Wark [15] for bubbles but are directly applicable to pendant drops.

Fritz summarized the calculations on the maximum volume (V_{max}), which occurs at a radius slightly under the maximum, by plotting V_{max}/a^3 against the angle θ. The maximum volumes (the greatest which will occur) are the highest points on the curves of Fig. 19 and a is the Laplace constant;

$$a = \sqrt{\frac{2H_{gl}}{(\delta_1 - \delta_g)}}; \; a = b \sqrt{\frac{2}{-\beta}} \quad . . . . (12)$$

where b is the radius of curvature of the lowest point of the drop (pt. 0 in Fig. 20). Magnitudes of V'_{max}/a^3 and $2x'_{max}/a$ were obtained from Wark, where V'_{max} is the maximum volume of drops without re-entrant angles and x'_{max} is the corresponding radius of the circle of contact. These results are plotted in Fig. 21 and tabulated in Table 13. From these entries it is seen that V'_{max} differs but little from V_{max}.

In order to illustrate the application of these data to water drops at 15, 40 and 70 C, Table 15 has been prepared which represents the maximum volume

[13] Bashforth, F. and Adams, J. C., An Attempt to Test the Theories of Capillary Action, (*Cambridge University Press*, 1883.)
[14] Fritz, W., *Physikalische Zeitschrift*, 36, 379 (1935).
[15] Wark, I. W., *Journal of Physical Chemistry*, 37, 623 (1933).

Table 13—The Thermal Function for the Evaporative Cooling of Water Drops in Air

°F	$\int_{170}^{t} \phi\,dt$, Air Temperature = 50 F				
	20 Per Cent R.H.	40 Per Cent R.H.	60 Per Cent R.H.	80 Per Cent R.H.	100 Per Cent R.H.
170	0	0	0	0	0
165	0.0108	0.0108	0.0109	0.0109	0.0110
160	0.0227	0.0229	0.0230	0.0231	0.0233
155	0.0361	0.0363	0.0365	0.0367	0.0370
150	0.0509	0.0513	0.0516	0.0519	0.0523
145	0.0674	0.0679	0.0684	0.0689	0.0694
140	0.0858	0.0866	0.0872	0.0878	0.0886
135	0.1066	0.1075	0.1083	0.1092	0.1102
130	0.1298	0.1311	0.1322	0.1334	0.1347
125	0.1559	0.1575	0.1590	0.1606	0.1625
120	0.1853	0.1874	0.1893	0.1915	0.1937
115	0.2186	0.2213	0.2239	0.2268	0.2297
110	0.2563	0.2600	0.2634	0.2672	0.2711
105	0.2994	0.3041	0.3086	0.3136	0.3187
100	0.3485	0.3547	0.3607	0.3673	0.3740
95	0.4045	0.4127	0.4207	0.4299	0.4401
90	0.4691	0.4803	0.4905	0.5034	0.5133
85	0.5447	0.5597	0.5731	0.5910	0.6057
80	0.6335	0.6533	0.6712	0.6966	0.7181
75	0.7379	0.7653	0.7916	0.8260	0.8575
70	0.8635	0.9014	0.9407	0.9886	1.0385
65	1.0169	1.0698	1.1300	1.2044	1.2879
60	1.2074	1.2852	1.3806	1.5050	1.6519
57.5	1.3216	1.4210	1.5394	1.7032	1.9129
55	1.4539	1.5858	1.7352	1.9603	2.2620
52.5	1.6094	1.7860	1.9860	2.3287	....
50	1.7942	2.0289	2.3240	....	
47.5	2.0179	2.3371	2.8360		
45	2.3027	2.7879			
42.5	2.6977	....			

°F	$\int_{170}^{t} \phi\,dt$, Air Temperature = 70 F				
	20 Per Cent R.H.	40 Per Cent R.H.	60 Per Cent R.H.	80 Per Cent R.H.	100 Per Cent R.H.
170	0	0	0	0	0
165	0.0113	0.0114	0.0116	0.0117	0.0118
160	0.0239	0.0242	0.0245	0.0248	0.0256
155	0.0379	0.0384	0.0389	0.0395	0.0406
150	0.0536	0.0543	0.0550	0.0559	0.0573
145	0.0710	0.0720	0.0731	0.0744	0.0762
140	0.0905	0.0920	0.0935	0.0955	0.0975
135	0.1125	0.1145	0.1166	0.1190	0.1218
130	0.1373	0.1400	0.1429	0.1460	0.1498
125	0.1654	0.1691	0.1729	0.1771	0.1820
120	0.1973	0.2023	0.2074	0.2131	0.2194
115	0.2338	0.2404	0.2472	0.2548	0.2631
110	0.2757	0.2853	0.2936	0.3038	0.3149
105	0.3239	0.3367	0.3476	0.3616	0.3763
100	0.3802	0.3969	0.4123	0.4310	0.4515
95	0.4462	0.4675	0.4890	0.5166	0.5459
90	0.5238	0.5525	0.5836	0.6238	0.6685
85	0.6170	0.6573	0.7030	0.7614	0.8373
82.5	....				0.9521
80	0.7306	0.7879	0.8576	0.9506	1.1028
77.5	....	0.8663	0.9555	1.0804	1.3279
75	0.8714	0.9566	1.0742	1.2530	....
72.5	0.9552	1.0618	1.2200	1.4882	
70	1.0508	1.1878	1.4033	1.8272	
67.5	1.1627	1.3588	1.6406	....	
65	1.2975	1.5532	1.9956		
62.5	1.4603	1.8120	....		
60	1.6590	2.1830			
57.5	1.9133	....			
55	2.2670				
...	....				

TABLE 13—THE THERMAL FUNCTION FOR THE EVAPORATIVE COOLING OF WATER DROPS IN AIR (Continued)

t F	$\int_{170}^{t} \phi dt$, AIR TEMPERATURE = 90 F				
	20 Per Cent R.H.	40 Per Cent R.H.	60 Per Cent R.H.	80 Per Cent R.H.	100 Per Cent R.H.
170	0	0	0	0	0
165	0.0118	0.0120	0.0123	0.0127	0.0130
160	0.0249	0.0255	0.0262	0.0270	0.0277
155	0.0396	0.0406	0.0418	0.0431	0.0443
150	0.0561	0.0577	0.0594	0.0614	0.0633
145	0.0746	0.0770	0.0795	0.0824	0.0851
140	0.0956	0.0988	0.1025	0.1065	0.1106
135	0.1192	0.1237	0.1288	0.1344	0.1402
130	0.1460	0.1524	0.1593	0.1670	0.1752
125	0.1767	0.1855	0.1950	0.2057	0.2181
120	0.2122	0.2240	0.2372	0.2519	0.2693
115	0.2536	0.2692	0.2868	0.3075	0.3315
110	0.3019	0.3229	0.3468	0.3775	0.4127
105	0.3583	0.3869	0.4222	0.4677	0.5253
100	0.4257	0.4655	0.5186	0.5893	0.6934
97.5	...	...	...	0.6695	0.8153
95	0.5077	0.5645	0.6413	0.7745	0.9820
92.5	...	...	0.7264	0.9164	1.2172
90	0.6083	0.6941	0.8261	1.1096	...
87.5	...	...	0.9596	1.3766	
85	0.7367	0.8743	1.1459	...	
82.5	...	0.9964	1.4189		
80	0.9069	1.1522	...		
77.5	1.0146	1.3620			
75	1.1435	1.6860			
72.5	1.3060	...			
70	1.5217				
...	...				

t F	$\int_{170}^{t} \phi dt$, AIR TEMPERATURE = 110 F				
	20 Per Cent R.H.	40 Per Cent R.H.	60 Per Cent R.H.	80 Per Cent R.H.	100 Per Cent R.H.
170	0	0	0	0	0
165	0.0125	0.0131	0.0136	0.0144	0.0152
160	0.0265	0.0278	0.0291	0.0308	0.0327
155	0.0422	0.0444	0.0468	0.0497	0.0530
150	0.0599	0.0634	0.0671	0.0717	0.0768
145	0.0800	0.0852	0.0908	0.0976	0.1055
140	0.1030	0.1102	0.1184	0.1284	0.1404
135	0.1293	0.1391	0.1510	0.1655	0.1834
130	0.1596	0.1732	0.1902	0.2115	0.2339
125	0.1948	0.2141	0.2385	0.2691	0.3113
120	0.23606	0.2634	0.2992	0.3467	0.4269
117.5	...	...	...	0.3967	0.5066
115	0.2852	0.3226	0.3776	0.4600	0.6083
112.5	...	...	...	0.5425	0.7413
110	0.3443	0.3980	0.4854	0.6536	...
107.5	...	...	0.5586	0.8061	
105	0.4157	0.4990	0.6474	...	
102.5	...	0.5636	0.7683		
100	0.5057	0.6408	0.9373		
97.5	...	0.7351	...		
95	0.6221	0.8551			
92.5	0.6951	1.0281			
90	0.7813	1.3171			
87.5	0.8843	...			
85	1.0168				
82.5	1.1962				
80	1.4613				
...	...				

Table 12—Steady-State Values of the Quantity "ψ"

Drop diameter, mm	0.2	0.5	1	2	3	4	5	6
Steady-state ψ, sec^{-1}	74.0	26.7	11.98	5.23	3.22	2.21	1.622	1.237

(V_{max}), the corresponding diameter of an equivalent spherical drop and the diameter corresponding to a spherical form of a pendant drop the volume of which is V'_{max}, that is, a pendant drop without re-entrant contour. Table 15a includes magnitudes of Laplaces' constant.

The drop volumes tabulated are maxima. Actual volumes will be less due to the effects of rate of growth and vibration. Fig. 20 indicates that part of the liquid below the ideal plane of cleavage is restrained from separating and this factor will reduce the ideal (maxima) volumes by perhaps 15-20 per cent. However, the results shown in Table 16 will aid greatly in establishing the water drop sizes which exist in atmospheric towers.

Formation of Drops from Jets

A jet of liquid may be formed either in the discharge of an appropriate nozzle or in the free space below a solid body over which the liquid is flowing at an appreciable rate. Both of these mechanisms are important in engineering applications; they represent the same phenomenon to a different degree as far as the jet portion of the system is concerned.

A liquid jet is an unstable body whose dispersion into drops is a spontaneous process. A thin sheet is yet more unstable than a jet and will always collapse into that form under the action of capillary forces. The intriguing forms assumed by liquid jets when generated in different ways and subjected to interactions with different objects early attracted the attention of physicists, who recognized the action of physical law in the observed phenomena. The work of Savart,[17] who discussed the form and interaction of jets with an obstacle; the proper recognition of the role of surface tension in the behavior of jets by Plateau;[18] the early theory of Hagen[19] on the dispersion of jets into drops; the observations of Buff[20] on the process by which drops formed at the end of a jet; and the extensive experimental observations of Magnus[21, 22] on the action and form of jets under many varied conditions will be cited in this connection.

Fig. 22A is taken from the interesting set of illustrations in the second article by Magnus. The characteristic operation of the dispersion process of low and moderate jet velocities is there brought out. Of interest is the way in which the jet undulations increase in amplitude until the connecting thread

[17] Savart, F., Ueber die Beschaffenheit der durch kreisrunde Oeffnungen aus dunnen Wand stromenden Flussigkeitsstrahlen. (*Annalen der Physik*, 109, No. 451, 520, 1834.)
[18] Plateau, J., Recherches, expérimentales et theoriques sur les figures d'equilibre, d'une masse liquide sans pesanteur. (*Annales de Chimie*, 30, Ser. 3, 203, 1850.)
[19] Hagen, G., Ueber die Auflösung Flüssiger Zylinder in Tropfen. (*Annalen der Physik*, 156, 559, 1850.)
[20] Buff, H., Einige Bemerkungen über die Erscheinung der auflösung des flussigen Strahls in Tropfen. (*Annalen der Chemie und Pharmacie*, 78, 162, 1851.)
[21] Magnus, G., Hydraulische Untersuchengen—I. (*Annalen der Physik*, 171, 1, 1855.)
[22] Magnus, G., Hydraulische Untersuchengen—II. (*Annalen der Physik*, 182, 1, 1859.)

TABLE 14—ILLUSTRATIVE EXAMPLE

(Evaporative Cooling of a 5mm Water Drop Falling from Rest Through Still Air at a Temperature of 70F and 50 per cent Relative Humidity)

S FT	θ SEC	DYNAMICAL FUNCTION	DROP TEMPERATURES F		
0	0	0	150	125	100
0.25	0.125	0.054	136.2	117.4	96.4
0.50	0.178	0.0895	129.3	113.2	94.2
1.0	0.250	0.1475	120.6	107.4	91.0
1.5	0.309	0.201	113.9	102.8	88.4
2.0	0.357	0.250	108.8	99.2	86.6
2.5	0.399	0.294	104.8	96.3	84.7
3.0	0.439	0.339	101.3	93.6	83.2
4.0	0.512	0.4235	95.6	89.4	80.7
5.0	0.575	0.501	91.4	86.1	78.6
6.0	0.634	0.576	87.8	83.3	76.8
8.0	0.739	0.717	82.6	79.2	73.9
10.0	0.836	0.856	78.7	75.9	71.8

of liquid collapses and drops are formed, with small auxiliary drops appearing from the segmentation of the connecting cylinder of liquid. Because a drop which is newly formed does not possess the equilibrium contour, and also because of the impact effect in the formation of the new surface upon separation from the jet, there will result an oscillation in the drop outline about the equilibrium shape. Evidence of this oscillation in seen in Fig. 22A in which the drops are successively flattened and elongated in the direction of motion.

TABLE 15—CRITICAL FUNCTIONS FOR PENDANT DROPS

θ	$\dfrac{V_{max.}}{a^2}$	$\dfrac{V'_{max.}}{a^3}$	$\dfrac{2x'_{max.}}{a}$
80	0.840	0.701	1.036
90	1.196	1.018	1.305
100	1.641	1.425	1.580
110	2.185	1.921	1.885
120	2.832	2.50	2.20
130	3.536	3.16	2.56
140	4.303	3.88	2.93
150	5.084	4.62	3.31
160	5.855	5.38	3.87
170	6.463	...	...
175	6.629	...	...
180	6.700	...	...

TABLE 15A—LAPLACE CONSTANT FOR WATER

TEMPERATURE C	F	H_{gl} DYNES/CM	$(\delta_o - \delta_g)$ GM/CM^3	a CM	a^2 CM^2
15	59	73.5	0.998	0.387	0.0579
40	104	69.6	0.991	0.375	0.0528
70	168	64.4	0.977	0.358	0.0460

A consideration of the form assumed by a drop in motion relative to the surrounding medium is next of interest. The rate at which the oscillations attendant upon formation are damped out depends upon the rate of energy dissipation due to the viscous forces operating within the drop, the inertia forces accompanying the motion of the fluid particles about their center of gravity, the resistance to surface deformation offered by the forces of interfacial tension, and the distribution of the dynamic pressure on the drop surface due to its motion relative to the surrounding fluid. Both the acceleration and velocity of a drop with respect to the surrounding gas influence the relative magnitudes of these effects. It is doubtful that a drop in motion ever attains an absolutely fixed contour. But when it is travelling at the steady-state velocity· a reasonably constant outline comes about in accord with the equilibrium of the forces governing the motion of the center of gravity. The flattening of the leading surface shown in Fig. 22B is an important effect. The

TABLE 16—THE MAXIMUM VOLUMES OF WATER DROPS FORMED ON THE UNDERSIDE OF A WETTED SURFACE WHEN THE RATE OF FLOW IS NOT SUFFICIENT TO FORM A JET

CONTACT ANGLE θ	$t = 15$ C			$t = 40$ C			$t = 70$ C		
	V_{max} cm³	d_{max}. mm	d'_{max} mm	V_{max} cm³	d_{max} mm	d'_{max} mm	V_{max} cm³	d_{max} mm	d'_{max} mm
90	0.0693	5.09	4.82	0.0631	4.94	4.68	0.0504	4.71	4.46
100	0.0951	5.66	5.40	0.0867	5.48	5.21	0.0755	5.23	4.98
110	0.1267	6.23	5.96	0.1155	6.04	5.79	0.1006	5.76	5.52
120	0.1640	6.80	6.52	0.1496	6.59	6.32	0.1302	6.29	6.03
130	0.2046	7.31	7.12	0.1868	7.09	6.81	0.1640	6.76	6.50
140	0.249	7.71	7.44	0.227	7.56	7.29	0.1980	7.22	6.97
150	0.294	8.25	7.99	0.269	8.00	7.74	0.234	7.63	7.38
160	0.339	8.65	8.40	0.309	8.38	8.14	0.269	8.00	7.77
170	0.374	8.94	...	0.341	8.66	...	0.297	8.26	...
175	0.384	9.01	...	0.350	8.74	...	0.305	8.34	...

V_{max}. = Volume of largest drop (with re-entrant contour) formed in equilibrium for given contact angles.
d_{max}. = Diameter of spherical drop of volume V_{max}.
d'_{max}. = Diameter of spherical drop of volume V'_{max}. (without re-entrant contour).

extent of this flattening is a function of the Reynolds Number for the drop, and its existence causes the drop to travel more slowly in the steady-state than would a true water sphere of the same volume. Data indicating the magnitude of this retardation have been presented in Fig. 1.

Returning to the process of jet dispersion, the first adequate theory was set forth by Lord Rayleigh [23] in 1878. For capillary vibrations on a cylinder of non-viscous liquid enveloped in a medium whose presence had no effect upon the phenomenon studied, he demonstrated that the ratio of the wave length of the surface wave to the cylinder (jet) diameter most favorable to spontaneous dispersion was $\pi\sqrt{2}$ or 4.44. The velocity of wave propagation was also shown to be a fairly complicated function of the surface tension and the ratio of the wave length to the diameter. From this information, it is possible to accurately compute the volume of a drop formed in the dissolution of a given jet under the conditions assumed in the derivation. A limiting criterion to be imposed is that the velocity of the jet relative to the medium (air)

[23] Rayleigh, On the Instability of Jets. (*Proc. London Math. Society*, 10, 4, 1878.)

through which it is passing shall not be so high as to result in additional surface forces.

Scheuermann [24] found good agreement between the Rayleigh theory and experiment for falling water jets. He also presented a corrected version of the Weisbach [25] equation for the median contour (about which oscillations take place) of a falling stream by adding the effect of the surface work involved in the change of cross-sectional area along the stream. The application of his equation in the analysis of falling jets will be indicated; it reads

$$y = \frac{v_0^2}{2_g}\left[\left(\frac{x_0}{x}\right)^4 - 1\right] + \frac{2H_{gl}}{\delta_1 x_0}\left[\frac{x_0}{x} - 1\right] \quad \ldots \ldots \ldots \quad (13)$$

Here x_0 = radius of the jet at the reference plane,
   v_0 = velocity of the jet at the reference plane,
   y = distance along the jet from the reference plane,
   x = radius of the jet corresponding to distance "y,"
   H_{gl} = interfacial tension, and
   δ_1 = unit weight of the liquid.

In 1931 Weber [26] published his treatise on the theory of jet dispersion to accompany the experimental study of Haenlein [27] on the same problem. These two papers constitute the best quantitative treatment available at present. It was there shown that the dispersion process took place in several distinct stages, namely: dispersion into drops without the influence of the air, dispersion into drops under the influence of the air, dispersion into waves under the influence of the air, and, finally, dispersion by atomization.

The presence of the air through which a jet is flowing has no appreciable influence upon the jet form and dispersion in the region of low jet velocities. Capillary forces are then the only means effective in the formation of surface waves and subsequent dissolution into drops. This case is illustrated in Fig. 23a. So long as the initial disturbances are all uniform, this process for a given system is characterized by a constant period of time, $T;$ the interval which is required for a given element of the stream to leave the nozzle and be converted into drop form. Hence, a definite length of jet, L, along which dispersion takes place is required for each jet velocity, V, i.e.,

$$T = L/V = \text{constant} \quad \ldots \ldots \ldots \ldots \ldots \ldots \quad (14)$$

Section A-B in Fig. 24 represents this action.

Circumstances which disturb the uniformity of the initial disturbances may at times cause the jet to disperse into drops at several points simultaneously, so that L is to be interpreted as a mean value.

When the jet velocity is increased, the behavior noted above undergoes a change which may be partially explained by the influence of the air. The air forces act upon the jet surface as does the wind in passing over a level water

[24] Scheuermann, R., Uber die Gestalt und die Auflösung des Fallenden Flüssigkeitsstrahles. (*Annalen der Physik*, 60, Ser. 4, 233, 1919.)
[25] Weisbach, Die Experimental Hydraulik. (F. Vieweg & Sohn, *Braunschweig*, 1885, page 51.)
[26] Weber, C. Zum Zerfall eines flussigkeitsstrahles. (*Zeitschr. für ang. Math. und Mech.*, 11, 136, 1931.)
[27] Haenlein, A., Zerfall eines Flussigkeitsstrahles. (*Forschung des Ingenieurwesens*, 2, 139, 1931.)

surface. Since the air velocity is appreciably higher at the crest of a wave than in the constricted portions of the jet, pressure forces come into play which accelerate the dispersion of the jet. This action is shown in Fig. 23b, and the resulting change in the length along which dispersion takes place is section B-C of the graph in Fig. 24.

With a still further rise in the jet velocity, wave-like initial disturbances deflecting the jet to one side appear, presumably due to the increased influence of the air and vortex action within the nozzle. Fig. 23c illustrates the phenomenon, and its action results in a continuation of the curve B-C in Fig. 24. In the case of a liquid of low viscosity, such as water, the wave form is not smoothly defined and small particles are easily torn from the surface. On the other hand, dispersion into waves is very distinct for viscous liquids.

The final process of dispersion is atomization, section C-D in Fig. 24. In this state, the liquid emerging from the nozzle lacks a smooth contour and the length along which dispersion takes place is given to erratic fluctuations. The influence of the air is quite marked in surface friction which helps to bring about the final dissolution of the stream into many small particles in intimate and chaotic mixture with the air.

The transitions between the individual jet forms take place gradually, so that, for example, a dispersion into drops and into waves, or wave formation and atomization, may occur as superimposed processes. The range of operating conditions in which each type of dispersion is most likely to occur has been determined by Ohnesorge.[28]

The analytical attack of Weber on dispersion phenomena succeeds very well in predicting the observed magnitudes and behavior. An extension of the Rayleigh analysis for viscous liquids undergoing dispersion into drops without the influence of the air leads to the expression

$$l/d = \pi \sqrt{2} \sqrt{1 + \frac{3\mu}{\sqrt{\rho H_{gl}d}}} \qquad \ldots \ldots \ldots \ldots \quad (15)$$

where μ is the absolute viscosity of the liquid and the other symbols are as previously defined. For a water jet of 0.51 mm diameter the observed values of l/d vary from 4.3 to 7.0 according to Haenlein,[29] while the predicted ratio is 4.47.

The time of dispersion, T, for a horizontal jet without the influence of the air may be represented by the equation

$$T \left(\frac{H_{gl}^2 \rho}{27\mu^3}\right) = 12 \left[\left(\frac{H_{gl}\rho d}{9\mu^2}\right)^{3/2} + \left(\frac{H_{gl}\rho d}{9\mu^2}\right)\right] \qquad \ldots \ldots \ldots \quad (16)$$

This relation may be applied as a first approximation to jets falling vertically if the liquid is not so viscous as to draw out into a long thread. The proper jet velocity and diameter to use here are then mean values. In combination with Scheuermann's correction of the Weisbach equation and a known value for the jet velocity and initial size, allowing the mean velocity and diameter to be determined, the equation permits an estimate of the length along which dispersion takes place and of the volume of the drops formed. The neat and

[28] Ohnesorge, W., Die Bildung von Tropfen an Düsen die Auflösung flüssiger Strahlen. (Zeitschr. für ang. Math. und Mech., 16, 355, 1936.)
[29] Loc. Cit. Note 27.
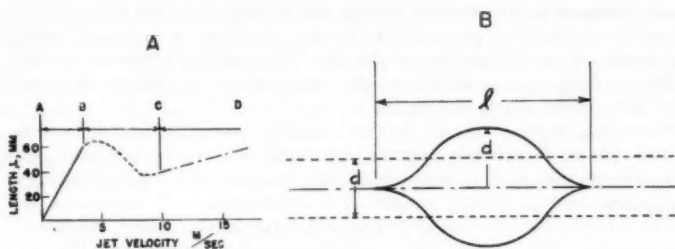
FIG. 24.   *A*. LENGTH ALONG WHICH DISPERSION TAKES PLACE FOR A HORI-
ZONTAL JET OF WATER 0.51 MM IN DIAMETER (HAENLEIN)

A–B   Dispersion without the influence of air
B–C   Dispersion under the influence of air
C–D   Atomization

B. DIAGRAMMATIC SKETCH OF COSINE WAVE FORM

rigorous solution of Weber may be applied to the determination of the outline
of the jet along its entire length when directed horizontally (and as an approxi-
mate solution for the vertical system), and the volumes of drops which would
be formed upon dissolution are thereby indicated. But, for an accuracy worthy
of the method it would be necessary to represent the outline by a Fourier
series.   A satisfactory practical estimate may be made by means of the
equation given, together with the assumption that the drops formed are solids
of revolution generated by revolving a cosine wave about the centerline of
the jet with the amplitude of the wave equal to the radius of the jet (Fig. 24B).
The volume of such a drop is $V = 3/8\pi d^2 l$, and since the ratio l/d is essentially
constant for any given conditions, it will be useful to employ the form

$$V = 3/8 \left(\frac{l}{d}\right) \pi d^3.$$

The diameter of a truly spherical drop of this volume is

$$d_s = d \sqrt[3]{\frac{9}{4}\frac{l}{d}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (17)$$

For $l/d = 4.47,$

$d_s = 2.20\ d_{jet}.$

The jet diameter to be used in this equation is the magnitude at the end
where dispersion is taking place.

Drop Formation in the Discharge of a Film-forming Pressure Nozzle

When it is desired to break up a stream of liquid into many small particles
at a fairly high rate, for the purpose of bringing these particles into well
distributed and violently agitated contact with a surrounding gaseous medium
so that an interphase exchange of material and energy may take place effec-
tively, some type of nozzle appropriate to the equipment involved is commonly
employed.   A simple non-clogging type of pressure nozzle is used most com-

monly for liquids which are not too viscous. The discharge of these nozzles is in the form of a cone-shaped film of liquid which quickly disintegrates into drops under the action of frictional and capillary forces causing instability.

The performance of a pressure-type nozzle is quite uniform in that the successive particles of liquid entering the nozzle undergo an essentially similar series of energy transformations. The energy of positions, velocity, and pressure possessed by the stream entering the nozzle is partly consumed in the viscous deformation, turbulence, friction, and increase in kinetic energy as the liquid passes through the nozzle, is further consumed by similar agencies acting within and upon the discharge film and drops, serves as the source of the work required in the creation of an extended liquid-gas interfacial surface, and then what remains appears in the energy of velocity and position possessed by the many drops formed. The influences which in particular determine the actions attendant upon the dissolution of the film are to be given attention. The behavior is similar to that of a single jet undergoing dispersion into waves and drops under the influence of the air, with atomization as the limiting phenomenon. Differences enter due to the manner in which the viscous forces resist the rapid spreading into a thin film and in the relative amount of capillary work involved in the creation of the very extended surface. Existence of this large surface is, of course, important for the transfer processes to be brought about and is the reason for employing a nozzle.

The extent of the film created and the size of the drops formed are determined by criteria of dynamic stability. A film in rapid motion relative to the enveloping medium is acted upon by surface forces which serve to accentuate any wave-like irregularities and to tear off by frictional action many small particles. Irregularities in the nozzle contour and vortex action within the nozzle determine the degree of chaotic motion and irregular form possessed by the film leaving the nozzle; at high discharge velocities they are predominant in destroying the film as such entirely, leading to atomization in a confused and turbid jet-like action. The capillary forces act in opposition to the extension of the film and are responsible for the appearance of waves therein and the collapse of the outer fringe of the film into jets during the dispersion process at low velocities. At higher velocities the frictional forces on both sides of the film and the whipping action of the outer fringe occasioned by the waves formed are favorable to small segments of liquid being curled back and torn off to ultimately collapse into drops. Oscillation of the drops so formed occurs about their equilibrium contour just as in the case of jets. The maximum size of a stable drop which may be formed in this way is determined by the relative velocity between the liquid particle and the surrounding medium (air in most cases) as noted earlier in this paper.

In a study of spray drying, Folger and Kleinschmidt [30] obtained photographs at intervals of 1/1500 sec. Collapse of the initial film segment into a ligament and then into a drop is evident. The action shown here is that of a liquid-solid mixture which is being dried in the process by evaporation of the liquid, so that the round particles finally formed are hollow. A similar behavior for a pure liquid to form a solid drop is indicated, however. It is important to recognize that the segment which is brought together by the action of the capillary forces is being acted upon by dynamic pressure forces of a con-

[30] Folger, B. B., and Kleinschmidt, R. V., Spray Drying. (*Industrial and Engineering Chemistry*, 30, 1372, 1938.)

siderable magnitude, so that the final unity accomplished is a tribute to the effectiveness with which the creation of an extended surface is resisted.

Folger and Kleinschmidt also present photographs of jets in which a spin chamber is located immediately behind the discharge orifice. If the nozzle pressure is insufficient to create an effective swirl in the spin chamber a non-expanding single jet is formed and the dispersion is similar to that reproduced in Fig. 22A. As the pressure is increased the hollow cone of the spray film appears which cone is finally ruptured due to the attenuation resulting from the combined effects of increasing cone diameter and the wave action at the liquid-gas interface.

A lower liquid viscosity results in a more tenuous film and allows the particles formed to assume a spherical shape more rapidly.

Conclusion

Effective design and operation of spray nozzles requires simple construction yielding a high effectiveness of energy conversion, with liquid particles of the proper size distributed throughout the reaction chamber. The behavior of drops may be predicted from the results presented herein. The results are to be considered as tentative until checked by experiment.

The ultimate goal of these researches is to present data which will allow the execution of a penetrating design of heat and material transfer equipment in terms of the predicted behavior of the small but discreet component systems whose superimposed behavior ultimately governs the macroscopic performance of the complete unit.[31]

Acknowledgment

The authors wish to acknowledge the interest exhibited in this project by the Southern California and Golden Gate Chapters of the ASHVE and the Air Conditioning Society of San Francisco.

Nomenclature

Consistent units must be used throughout.

A = projected area of drop, based on the diameter of the equivalent spherical volume.

$\bar{a}$ = thermal diffusivity = $\frac{k}{c_p \delta}$.

a = Laplace constant = $\sqrt{\frac{2Hg_1}{(\delta_1 - \delta_g)}}$

b = radius of curvature of drop at lowest point (0 in Fig. 20).
C = aerodynamic resistance coefficient.
c_p = unit heat capacity at constant pressure.
d = equivalent drop diameter, or cylinder diameter.
d = differential operator.
f = unit thermal conductance.
g = gravitational constant.
H_{gl} = interfacial liquid-gas free energy per unit area (surface tension).
k = thermal conductivity.
m = mass of drop.

[31] Nottage, H. B., Master of Science Thesis entitled, On the Evaporative Cooling of Water Drops, with a Consideration of Drop Dynamics. (On file, *Library*, University of California, Berkeley.)

P_e = partial pressure of vapor in air far away from the drop surface.
P_w = partial pressure of vapor in equilibrium with the liquid at temperature "t".
r = latent heat of evaporation at temperature "t".
R_v = gas constant of vapor.
R = equivalent radius of a spherical drop.
S = distance of fall or rise.
t = temperature of drop.
t_a = temperature of air-vapor mixture far away.
T_m = absolute temperature corresponding to arithmetic mean of "t" and "t_a".
U' = over-all unit mass conductance based upon concentration potential.
V = volume of pendant drop.
v = drop velocity.
x = radius of horizontal section through a pendant drop at the plane of separation.
$\bar{\beta}$ = coefficient of thermal expansion.
δ = weight density of fluid.
θ = time interval.
μ = absolute viscosity of fluid.
ρ = mass density of fluid.
σ = one of the principal radii of curvature at point A (Fig. 20).
ϕ = angle of contact at point A (Fig. 20).
Φ = coefficient of the thermal function, Equation (10).
ψ = coefficient of the dynamical function, Equation (9).

Dimensionless Moduli

β = drop force modulus = $-\dfrac{b^2(\delta_1 - \delta_e)}{H_{el}}$

Pr = Prandtl's modulus = $\dfrac{\mu c_p g}{k}$

Fo = Fourier's modulus = $\dfrac{\bar{a}\theta}{R^2}$

Gr = Grashof's modulus = $\dfrac{d^3\rho^2 g\bar{\beta}\,(t - t_a)}{\mu^2}$

Nu = Nusselt's modulus = $\dfrac{f_c d}{k}$

Re = Reynolds' modulus = $\dfrac{v d \rho}{\mu}$

Maximum volume modulus = $\dfrac{V_{max}}{a^3}$ or $\dfrac{V'_{max}}{a^3}$, the latter referring to drops without re-entrant contours.

Subscripts

$\rightarrow$ = relative to.
0 = referring to initial condition.
c = convective heat transfer.
r = radiant energy transfer.
a = air-water vapor mixture far away from drop.
l = liquid (water).
g = gas.

Note: Symbols for Equations (13) to (17), inclusive, are defined in the text following these equations.

DISCUSSION

S. R. LEWIS: The University of California should be complimented for presenting the results of a very remarkable fundamental research. We have suffered for lack of this very information. I get tired of cooling tower people coming around with their private and confidential information which is proprietary, and I do not think they know very much about it anyway. We are on the way to find out something, with thanks to the University of California.

H. B. NOTTAGE: We certainly appreciate your encouragement. With the comments of members of the Society to guide us and continued support of our research by the Society we are sure that we can do more of the things which all of us would like to see done.

C. H. CHALMERS: Does the fact that we have hail drops out in Minnesota as large as hen's eggs, offer any contradictory evidence in the matter of the size of rain drops?

MR. NOTTAGE: The question is whether or not very large hailstones behave as rain drops. I would not call them rain drops. Hail is ice. Hailstones grow continually as they pass through the air and freeze out additional moisture. A solid body cannot be dispersed by a dynamic fluid pressure unless it is subjected to a terrific impact. Whereas, if a liquid body be subjected to even a constant dynamic pressure it will be somewhat deformed, and if certain limiting conditions of stability are exceeded a shattering of the single body into many smaller ones will follow. The deformation noted does not come about in the case of solid hailstones and the data given for liquid water do not apply. We could, however, study the behavior of hailstones by methods similar to those used for drops.

I am prepared to substantiate my statement on the maximum size possible for a stable drop with actual data. Observations on the size of rain drops have shown that we do not have rain drops larger than roughly seven millimeters in diameter, and that size is only occasional in a very heavy rain.

W. H. DRISCOLL: I feel that everyone has been inspired by this paper. It is one of those things that appears before us occasionally and that none of us understand but, as our knowledge of the particular problem increases with the thought that we give to the subject and as we apply the facts presented in the paper to the practical problems with which we must contend, we grow up to a better understanding of the subject. This paper, like many papers that are presented before this Society, was prepared as a result of long hours, and perhaps months of research. The result is that its contents cannot possibly be grasped in the few moments of its presentation by an audience, no matter how intelligent that audience may be. I am sure that all will agree that it is a very worthwhile paper, a paper that will prove of great value, and it is a splendid contribution to the literature of the Society.

MR. MOORE (Washington): Has any method for measuring the size of drops in an actual installation been developed?

MR. NOTTAGE: We have developed methods for measuring the size of small drops. The apparatus has been described briefly in the previous paper [14] presented by Professor Boelter. We are also working on a photographic equipment to enable us to measure the size of actual drops formed in, for example, the interdeck spaces of atmospheric type cooling towers. We can photograph the drops and also catch them and weigh them to determine the actual drop sizes. Observations though, indicate that the range of data which I have given here is sufficient to cover the usual conditions existing in evaporative cooling equipment except in the case of a very finely atomized liquid. That problem has been taken up by Professor Johnstone at the University of Illinois. He has dealt with very, very small drops and he has developed a method of predicting volume transfer coefficients which includes the probability of coalescence of these small drops.

If you have two drops in motion and they come together they will normally coalesce. That is because the total surface area for the single drop of increased volume is less than it was for the two initial drops taken separately, and surface area represents a certain amount of energy stored in the system. A spontaneous decrease in the amount of energy stored within a system will always come about when possible,

[14] Loc. Cit.   Note 1.

so a decrease in the total liquid surface area by the union (perhaps only momentary) of two or more drops upon collision is a logical expectation.

Then, of course, if the limits of stability for the enlarged drop should be at any instant exceeded, as might come about in the impact effect of a jet of air striking the drop or as a result of accelerated motion under the action of gravity, a breaking up into smaller drops once more will follow. In actual forced draft towers, these processes of successive coalescence and disintegration comes about to a considerable extent in the free spaces. We have studied this action by means of a stroboscope as well as by direct visual observations, and we expect to understand it more completely in due time.

W. L. Fleisher: This paper is the second of a series of papers from the University of California sponsored by the Committee on Research of the Society.

In a way the abstractness of this paper worried the Committee as to the possibilities of its presentation. All of us who saw the exactness with which they were carrying on their work in California were so impressed that we felt that it was one of the vital papers to be presented to the Society.

In reply to Mr. Driscoll, the chairman of the Committee on Research had to read this paper and approve it before publication.

Mr. Chalmers: I am very much interested in what the author had to say about these jets. There are some two million homes in America heated with oil. A great many, possibly half, have what we call pressure-type burners in which we have a nozzle and a jet where the oil comes out and mixes with the air, the oil jet being in the center surrounded by a tube which gives—possibly you might call it a jet of air.

We have many problems there, and one is to get quietness, which carries us back to fine atomization. Then we have the problem, in order to secure high efficiency, of mixing the two jets. I wonder if you have done any work at all along the lines of spraying oil and spraying air and getting them adequately mixed together so that we may get the highest possible efficiency.

You possibly know that in oil burner trade literature they sometimes show the air moving in a circular motion as it comes out, and the oil they show moving in an opposite circular track. Some of us doubt whether the oil does so move or not.

Have you done anything along that line, with the idea of helping us oil burner engineers to get high CO_2?

Mr. Nottage: We have not done anything definite ourselves. As I recall though, there have been articles published on oil burners in such magazines as the *ASME Transactions*. Regarding the possibility of undertaking detailed studies, all I can say at present is that the dispersion of a body of oil conforms to physical law according to the same basic principles as does the dispersion of a body of water, so we could apply these principles to the analysis of the behavior of oil burners. It is merely the general problem of dynamical interaction between a liquid and a gas that is the subject for study. The principles are fundamentally the same but the details introduced in carrying out an exact analysis become at times rather involved, although the promise of reward is indeed rich in most instances.

Mr. Lewis: Is there any probability that the question of the noise from cooling towers will be taken up in connection with this study?

Mr. Nottage: We have recognized the problem of noise. You may have read the paper [18] previously presented by Professor Boelter in which we listed the variables

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[18] Loc. Cit. Note 14.

which we visualize as important in the operation of cooling towers. Noise was one of those variables. We have heard complaints about noise but we have not as yet undertaken to set forth specific recommendations. However, we have a certain insight to guide our thoughts in that noise is caused, if one excludes the operation of machinery, by the action of nozzles and fans and impact of liquid jets and drops on solid surfaces. If we can learn to build towers such that the impact effects will be reduced to a minimum, and if quieter nozzles and fans can be employed, the noise level may be reduced below an objectionable value.

Mr. Lewis: There seems to be considerable question among cooling tower people as to whether the fans should induce or should force the air through. I imagine that we will, before we finish, find more about the relative efficiencies of the two methods.

We had one case of a ten story building in which a cooling tower was placed at the third story level. When the cooling tower was in operation, all of the windows on that side of the building had to be closed due to noise, since the tower was of the induced draft type. We were called in to try to find some way to overcome this. Well, we changed the fans so as to blow in a horizontal direction to deliver the air and noise out over the neighbors, who were not quite so important as those in the tall building, and I think we got away with it.

Mr. Nottage: As to the question of the comparative value of induced or forced draft, I have no specific data to submit on the point at present. The one aspect of the question is that in the case of an induced draft tower the total pressure prevailing within the system is somewhat less than that in the case of a forced draft tower.

It may be well to recall in that connection the classical experiment of elementary physics whereby water is made to freeze by a reduction of the pressure over the liquid to a point where cooling by evaporation is extremely rapid. The same principle would seem to have application to the cooling tower, where the bringing about of the lowest practical total pressure would certainly be favorable to an increased cooling effect.

I cannot specify the exact magnitude of the improvement in performance which would be feasible in an actual case. The problem is one which awaits further decisive investigation.

HEAT GAIN THROUGH GLASS BLOCKS BY SOLAR RADIATION AND TRANSMITTANCE

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Pittsburgh, Pa.

THE growing use of glass block in buildings provided with summer cooling and air conditioning has made it increasingly desirable that acceptable design data on heat gain through such constructions be established. There has been emanating from the field such diverse information, that, even though some of it came from accredited research sources, it was deemed expedient that the Research Laboratory of the American Society of Heating and Ventilating Engineers, in the light of its disinterested position in the industry, make independent research to establish data on which both the public and the manufacturers could rely, completely and authoritatively. Having in the Society representatives not only from the technical schools of the country, but also from the manufacturing, engineering and contracting groups, organized into a technical advisory committee to investigate any subjects pertaining to the use of glass in air conditioning buildings, the investigation of this matter at this time is proper and logical. Consequently, exhaustive research was started at the Research Laboratory in Pittsburgh with the latest and most complete equipment, from which the results indicated in this paper emanated.

The study was made in a two-room test house, Figs. 1 and 2, built for the purpose on the roof of the Bureau of Mines warehouse building. The design of the rooms and the test procedure were similar in many respects to those used in an earlier Laboratory study of sun radiation gain through windows.[1] Each room was about 7 ft x 7 ft x 8 ft high inside and was designed so as to provide for the insertion of a 4 ft x 6 ft window panel in one side. The test house was built on a turntable so that these windows could be made to face in any direction. The walls of each room were well insulated, having an air to air transmittance coefficient of 0.06 Btu per square foot per hour per degree Fahrenheit. They were painted aluminum on the outside to minimize the absorption

of solar radiation, and in addition the two sides and the top surface were shaded from sun radiation by canvas placed on a frame-work about 1 ft from these surfaces. The inside walls, ceilings, and floors were painted white. Black cheesecloth screens placed within the rooms intercepted direct radiation from the sun so as to make this heat gain immediately effective in raising the air temperature of the rooms and therefore in the measured cooling load.

Although in the construction of the test house all the joints were made tight and then sealed in order to reduce infiltration to a negligible value, tests were made by the carbon dioxide dilution method under different weather conditions to determine the rate of infiltration for each room. It was found
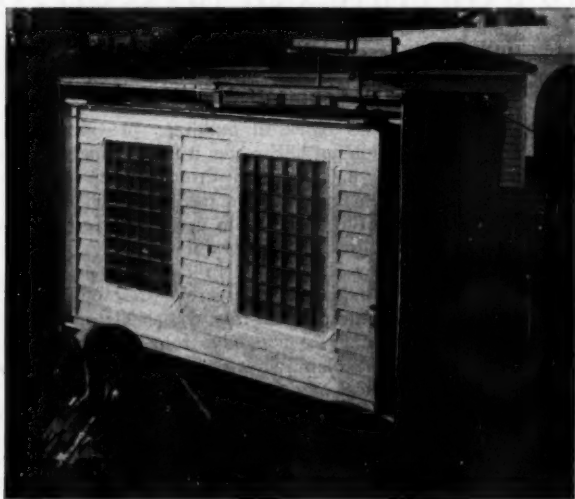
Fig. 1.   Test House with Glass Block Panels in Position
for Test

that this rate did not exceed 0.03 air changes per hour. With the maximum outside to inside air temperature difference observed for any test, this gave a total heat gain for one room of 4.5 Btu per hour or 0.19 Btu per square foot of glass.

Each room was provided with a cooling unit, modified as indicated in Fig. 3 so that a constant volume of air was taken from within 4 in. of the floor, passed over the ice or, around it through a by-pass, over the electric heaters, through an auxiliary fan and then dispersed into the upper part of the room through a conically shaped cheesecloth bag with air velocities not exceeding 38 fpm at any point within 6 in. of any wall surface. The rate of cooling was controlled by by-pass dampers operated from the outside of the room. The drip from the melting ice was piped to weighing buckets in the observation room and weighed at 20 min intervals. A thermostat within each room controlled a small electric heater, located so as not to affect the ice
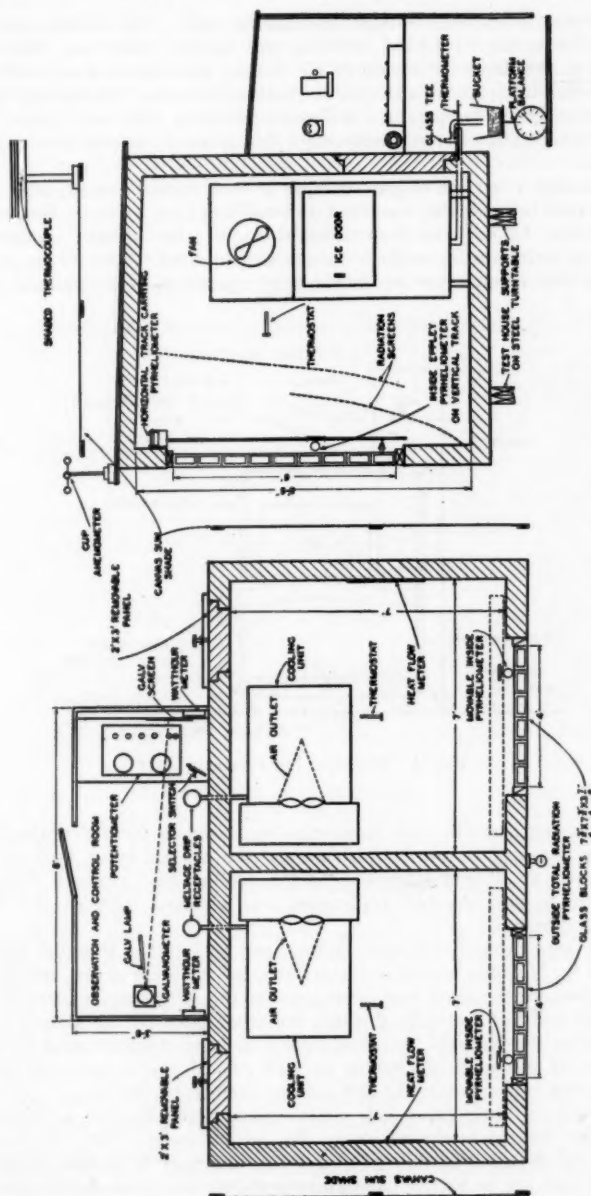
FIG. 2. TEST SET-UP FOR DETERMINING SOLAR RADIATION THROUGH WINDOWS

meltage, in the air stream through the cooling unit. This arrangement gave a lateral distribution of cooled air near the ceiling which was then drawn downward to the floor. The rate of air change was adjusted to 24,400 cu ft per hour which was estimated to give the most satisfactory relationship between air velocity and air temperature difference between inlet and outlet. With maximum heat gain on hot sunny days this gave a temperature difference of 5.2 deg.

In conducting a test the by-pass damper in each cooler was adjusted to give a cooling rate a little greater than the estimated heat gain. The heating capacity of the electric heater was then adjusted to a value a little greater than necessary to maintain the uniform temperature desired. The thermostat then maintained this temperature by intermittent operation of an electric heater.
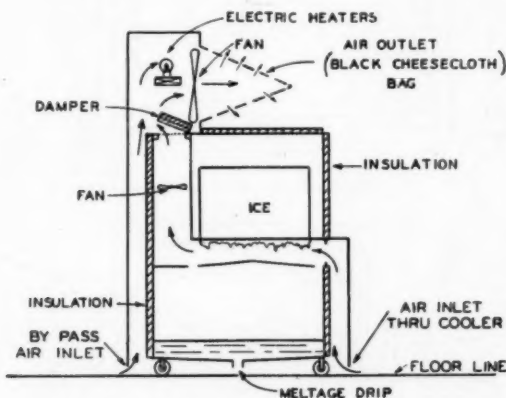
Fig. 3.   Melting Ice Cooling Unit

The total cooling load for each room was then taken as the difference between the cooling measured by the ice meltage and the electric input, both of which were observed at 20 min intervals. As the cooling load changed throughout the day, the by-pass damper adjustment was changed between the 20 min observation intervals.

The heat gain through the floor, ceiling, and walls other than the glass was determined by Nicholls heat flow meters attached to these areas, and the total heat gain through the glass was determined as the difference between the total cooling load and the heat gain through the other parts of the structure. The emf's given by the Nicholls heat flow meters, and the thermocouples giving the temperature of the air at the center of each cubical, the temperature gradient throughout the room, the inside and outside surface temperatures, and the air temperature 6 in. from either side of the glass were observed by a precision potentiometer located in the adjoining observation room. The solar radiation normal to the direction of the sun's rays was observed by potentiometer readings of the emf set up by the Laboratory pyrheliometer described in an earlier

Laboratory publication.[2]   The sun radiation intensity impinging against the outside vertical glass surface was observed by a low sensitivity type Eppley pyrheliometer attached to the wall of the building between the two windows.   A second high sensitivity type Eppley pyrheliometer located inside of each room, with its sensitive plane parallel to the inside surface of the glass panel, and moved about so as to traverse a 3 x 3 ft area was used to measure the amount of radiation passing through the glass panel.   This was accomplished by mounting the pyrheliometer on a vertical carriage to give it the vertical translation; this carriage, in turn, being suspended from a horizontal carriage to give it the horizontal translation.   The speed of these two movements was then regulated so as to give an approximately constant pyrheliometer reading.   The Eppley instruments have plane sensitive elements so located as to have a view

TABLE 1—DESCRIPTION OF GLASS BLOCKS STUDIED AND DESIGN HEAT GAIN DATA

BLOCK TYPE	OUTSIDE SURFACE	INSIDE SURFACE	BLOCK AIR SPACE SURFACES		DESIGN HEAT GAIN FOR AUG. 1 FOR A SOUTH EXPOSURE AT 41 DEG N LAT. (IN BTU PER SQ FT PER HOUR)			
					Radiation		Total	
			Outer	Inner	Max. Per Hour	Per Day [a]	Max. Per Hour	Per Day [a]
A	Smooth	Smooth	Vertical Ribs	Horizontal Ribs	22.0	115.5	40.5	250.4
B	Vertical Ribs	Vertical Ribs	Smooth	Smooth	19.8	100.3	38.4	233.6
C	Ribs on one or both surfaces		Horizontal Prismatic Ribs	Horizontal Prismatic Ribs	17.0	91.8	35.0	216.8

[a] 9 A.M. to 5 P.M.

of the entire hemispherical angle or a solid angle of π radians, unobstructed by any throat or aperture through which the sun must shine as is the case with the ASHVE Laboratory and the Abbot Silver Disk instruments.   They have their sensitive elements encased in a thin glass dome which absorbs a certain percentage of the solar radiation impinging on the outside.   This absorption is selective and includes a larger percentage of energy in the long waves, the glass being opaque to practically the entire radiation from a black body temperature source below 400 F.   The calibration of the instrument is naturally corrected for that part of solar radiation which is absorbed.   However, when the instrument is used to measure radiation from lower temperature sources a considerable error results.   As an example, the use of these instruments inside the rooms measures satisfactorily that portion of radiation which came through the glass, but as should be expected they were not at all sensitive to re-radiation from the inside surface of the glass block, which must have been considerable because of the elevated temperature resulting from

[2] ASHVE RESEARCH REPORT No. 923. Heat Transmission as Influenced by Heat Capacity and Solar Radiation, by F. C. Houghten, J. L. Blackshaw, E. M. Pugh and Paul McDermott. (ASHVE TRANSACTIONS, Vol. 38, 1932, p. 231.)

the radiation absorption. The Eppley pyrheliometer observations were made with the precision type potentiometer in the observation room. The wind velocity was observed by a cup-type anemometer and the outside shade temperatures were determined with a thermocouple located about 3 ft above the top of the building.

Six different designs of glass blocks were tested, three made by each of the two manufacturers. All of the blocks were 7¾ in. x 7¾ in. x 3⅞ in. which,
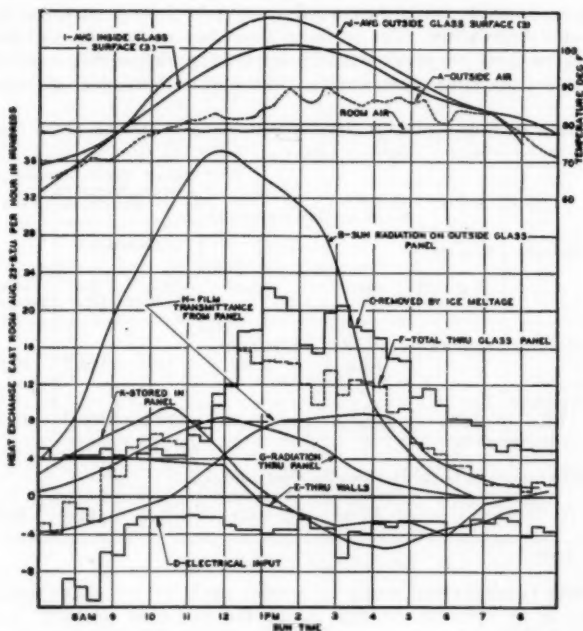
FIG. 4. TEST DATA FOR AUG. 23. EAST ROOM EQUIPPED WITH TYPE *A* GLASS BLOCK PANEL. ALL HEAT EXCHANGE RATES APPLY TO THE ENTIRE ROOM OR THE 24 SQ FT PANEL

with a ¼ in. mortar joint, filled an 8 in. x 8 in. wall area. Blocks of each design were made up into a 4 ft x 6 ft glass block panel in a wooden frame. This frame containing the glass panel could then be placed into the opening in either room, fastened in place, and then caulked with hair felt for insulation. The joint was taped on either side so as to eliminate air leakage. With this arrangement glass block panels could be made up and allowed to age before being put into place for test.

The six designs of glass block tested were chosen in pairs having certain common characteristics including one design from each manufacturer. The characteristics of the designs and key numbers used for the different blocks tested are given in Table 1. By agreement individual ratings of the several

blocks of the two manufactuers are not given, but rather the average performance for each of the three types.

Tests were run from 6 A.M to 9 P.M. during a majority of the days from June 26 until September 1 but the weather was such that only 15 of these days yielded acceptable data. Tests were also made in September from which additional sets of data were obtained.

Most of the tests were made with the glass block panels facing south. On a few occasions tests were run with the panels facing east or west, and tests were also run to give the effect of inside and outside shading of the panel
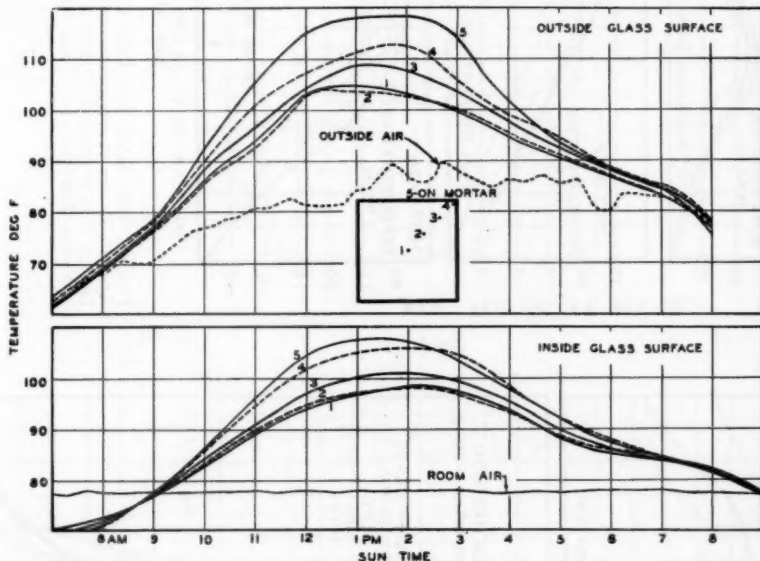
FIG. 5. TEST DATA FOR AUGUST 23 (EAST ROOM) GIVING INSIDE AND OUTSIDE SURFACE TEMPERATURES OF A REPRESENTATIVE GLASS BLOCK AND ADJOINING MORTAR. CURVES NO. 3 REPRESENT INTEGRATED AVERAGE FOR THE ENTIRE AREA

and to obtain a comparison between the heat gain through a glass block panel and single glazed steel sash of the same size.

As an example of the way the data were treated a record of the results for one glass panel during a test on August 23 when the glass was facing south is shown in Fig. 4. This was a fairly cool but bright sunny day. The outside air temperature, *Curve A*, reached the maximum of 90 F and the Pittsburgh Weather Bureau recorded a maximum of 87 F.

The sun radiation, *Curve B*, while somewhat distorted (rather high radiation during the middle of the afternoon compared with the forenoon), will be shown later to differ little from what has been assumed as design solar radiation for this time of the year. It was obtained by the Eppley pyrheliometer attached to the southside of the building, that is, *looking* south and is, therefore, a measure
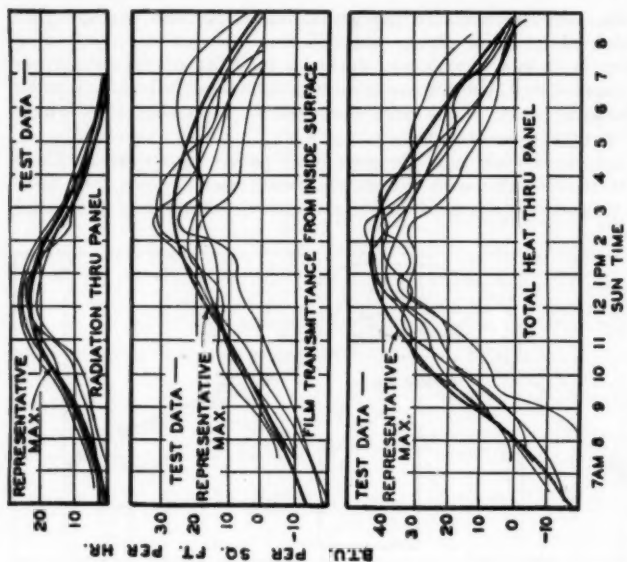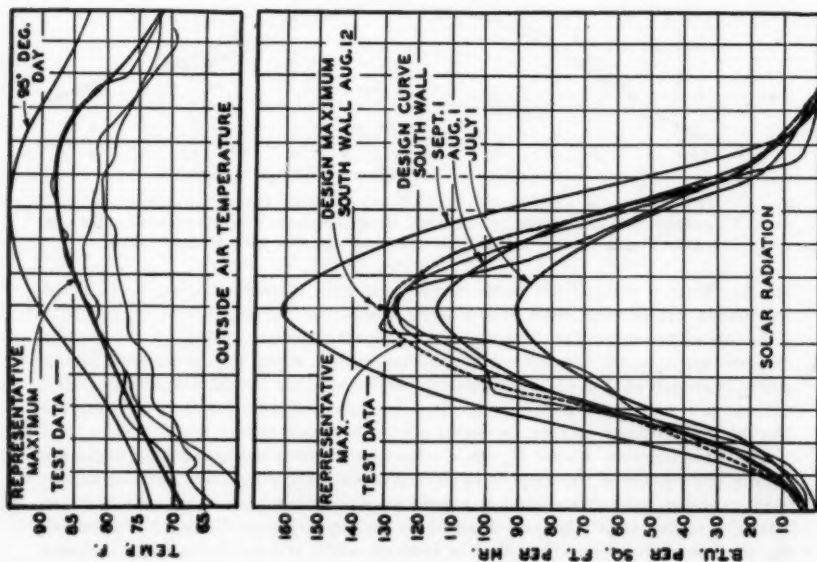
Fig. 6. Test Data for Two Type *A* Glass Blocks for Aug. 10, 11 and 14 with Representative Maximum Curves. Design Sun Radiation Curves for South Wall and Outside Air Temperature Curve for 95 F Day Also Given

of the solar radiation intensity normal to the south windows. Like all other heat rates in this chart, the radiation is expressed in Btu per hour for the entire 24 sq ft area of the window.

The rate of heat extraction by melting ice, *Curve C,* results from the actual meltage as measured at 20 min intervals, while the electrical heat input used to give thermostatic control is given by *Curve D.* The heat gain through the walls other than the windows as calculated from the Nicholls heat flow meter observations is given in *Curve E.* The total heat gain through the 24 sq ft glass block panel as shown in *Curve F* is the difference between the total heat removed by the melting ice, *C,* and the sum of the heat gain by electrical
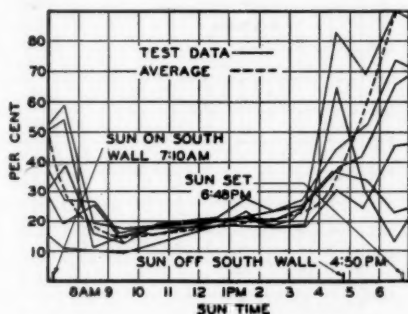
FIG. 7. RADIATION THROUGH GLASS PANEL EXPRESSED AS A PERCENTAGE OF THE TOTAL INCIDENT SOLAR RADIATION STRIKING THE OUTSIDE GLASS PANEL FOR TESTS ON GLASS BLOCK TYPE *A* SHOWN IN FIG. 6

input, *D,* and by transmission through the walls other than the windows, *E.* or $F = C - (D + E)$. *Curve G* is the radiation through the glass as measured by the Eppley pyrheliometer inside the room. The difference between the total heat gain through the glass and the radiation gain is given in *Curve H;* this is the gain by film transmittance from the inside surface of the glass. It should be noted that this represents the heat transfer from the inside surface to the inside air by radiation and convection due to the surface to air temperature difference, and does not include the radiation coming through the glass which was measured by the inside pyrheliometer. It will be observed that since H is determined by a difference, there will be crowded into it all errors which may have resulted in the observations and deductions used to obtain the several values cited above.

The inside and outside average glass surface temperatures are given by *Curves I* and *J,* respectively. It is of interest to note the high elevation of these temperatures above that of the inside or outside air. This elevation in temperature results, of course, from the absorption of solar radiation by the glass block panel including the mortar joints. *Curve K* gives the storage of heat within the glass panel as calculated from its average rise in temperature, specific heat, and weight. It is of interest to note that during the early

morning hours heat was being stored within the glass at a higher rate than the total heat gain to the room through the glass.

Variations in the inside and outside surface temperatures of the glass throughout the day are given in Fig. 5.   The small sketch shows the location of the thermocouples on the surface with respect to the surface dimensions of
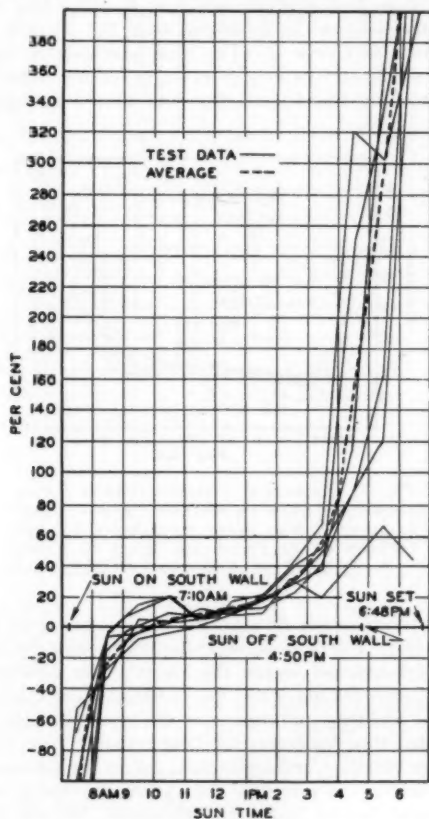
FIG. 8. FILM TRANSMITTANCE FROM INSIDE SURFACE EXPRESSED AS A PERCENTAGE OF THE TOTAL INCIDENT SOLAR RADIATION STRIKING THE OUTSIDE GLASS PANEL FOR TESTS ON GLASS BLOCK TYPE *A* SHOWN IN FIG. 6

the glass and mortar joints. As would be expected, the mortar joints, absorbing the greater percentage of incident radiation, reach a higher temperature. An integration of the temperature over the area of the block represented by each thermocouple shows that *Curve 3* represents a good average integrated temperature for the entire block and its percentage of the mortar joint. These

temperatures were used in *Curve I* and *J*, Fig. 4, for determining the rate of heat storage in the panel. Analyses similar to those shown in Figs. 4 and 5 were made for each test on each of the six types of glass panel and are summarized in subsequent figures.

The results of three fairly satisfactory tests on the *A* panels facing south are given in Fig. 6; showing from top to bottom, outside air temperatures,
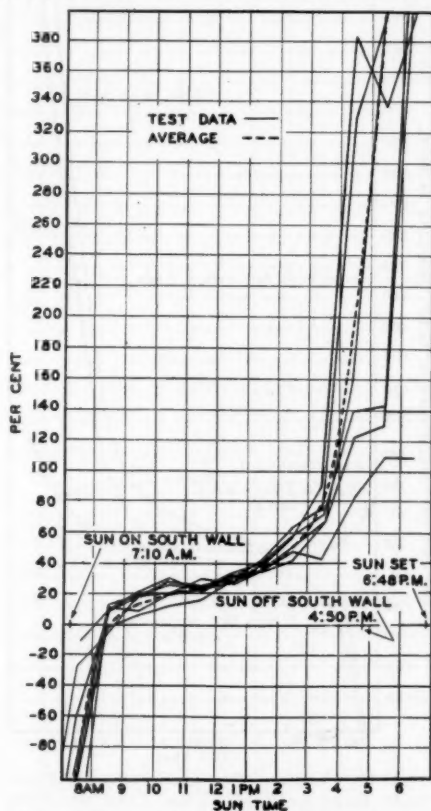
FIG. 9. TOTAL HEAT GAIN THROUGH PANEL EXPRESSED AS A PERCENTAGE OF THE TOTAL IN-CIDENT SOLAR RADIATION STRIKING THE OUT-SIDE GLASS PANEL FOR TESTS ON GLASS BLOCK TYPE *A* SHOWN IN FIG. 6

solar radiation intensities normal to the outside surface of the panels, radiation through the panels, the difference between the total gain and radiation gain or the inside film transmittance and finally the total heat gain. In each chart the light-weight curves represent the actual observed values for the individual tests on the two Type *A* panels for the three days, August 10, 11, and 14.

The heavy line curves are given as representative maximums more or less determined as the upper envelope of the curves representing the test data. These average maximum curves were considered as representing the effect of the maximum outside air temperature and the sun radiation observed for the
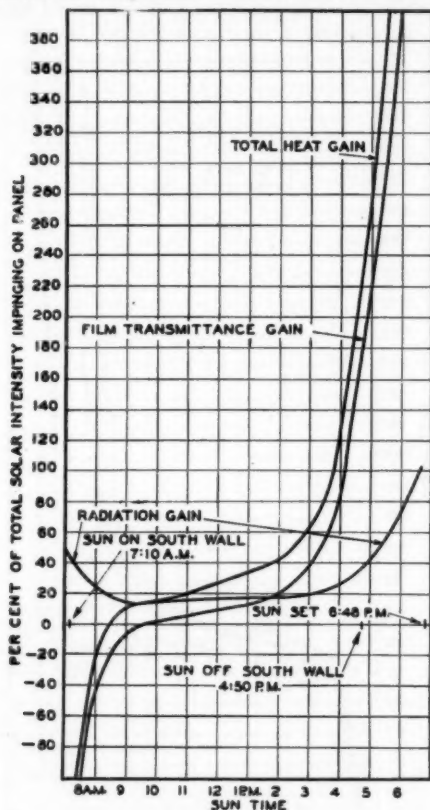
Fig. 10. Average Curves of Total Heat Gain Through Panel, Film Transmittance from Inside Surface of Panel and Radiation Gain Through Panel Expressed as a Percentage of the Total Solar Radiation Incident on the Outside Glass Panel Surface for Tests on Glass Block Type *A* (Summary of Figs. 7, 8 and 9)

several tests.   For comparison there are also plotted in this figure suggested design sun radiation curves for a south wall for July 1, August 1, and September 1.   It will be noted that the average curve used which may be assumed to apply on August 12 fits very well into the series of design curves.   The probable outside air temperature curve for a day in mid-summer when the
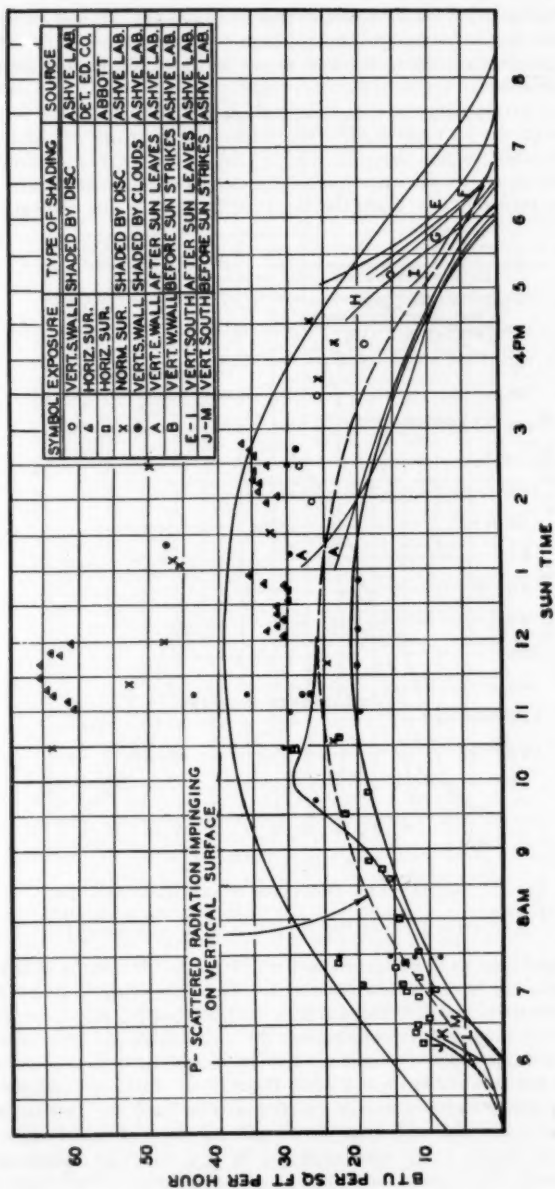
FIG. 11. SCATTERED SKY RADIATION FROM VARIOUS SOURCES AS INDICATED BY KEY

maximum reaches 95 F is also plotted for comparison. It will be noted that this curve which may be accepted as a design temperature curve for Pittsburgh and many other metropolitan districts shows considerably higher temperatures than those observed.

In order to extrapolate the data collected on any particular day or group of days to a design day as regards air temperature and intensity of solar radiation on the south wall, curves showing the percentage of solar radiation incident normal to the panel which was transmitted through as radiation are given in Fig. 7. Film transmittance from the inside glass surface and the sum of these
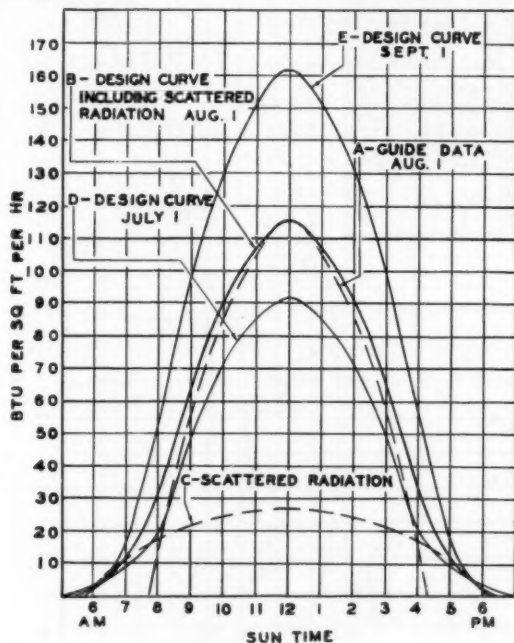
FIG. 12. DIRECT AND SCATTERED SOLAR RADIATION IM-
PINGING ON A SOUTH WALL AND RESULTING IN DESIGN
RADIATION CURVES FOR JULY 1, AUG. 1, AND SEPT. 1

two or the total heat gain are given in Figs. 8 and 9. It will be noted that the curves all take on a characteristic form. The approximate times at which the sun appeared on and left the panel as well as that of sunset are indicated. The portion of the curves representing the time when the sun was on the wall are of course most significant, and it should be pointed out that the scattering of the curves before and after these times while of academic interest is not of any great significance for the reason that both the rate of heat gain and the intensity of solar radiation were small for these periods. The negative values in Figs. 8 and 9 are accounted for by the fact that during the early morning periods the rooms lost heat to the outside. The high positive

percentages after the sun left the south wall are accounted for by the fact that, although heat gain by transmittance was still considerable in magnitude, the solar radiation intensity with which it was compared had become very small. As pointed out above, however, the percentage curves for the period of day
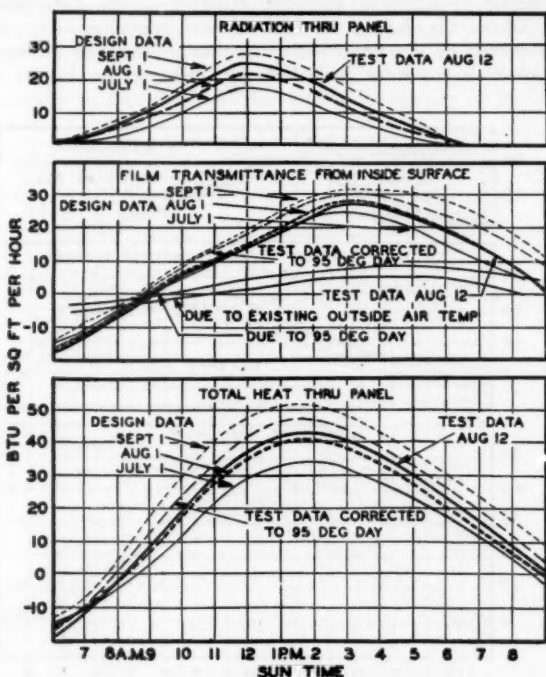
FIG. 13. JULY 1, AUG. 1 AND SEPT. 1 DESIGN DATA FOR TOTAL HEAT GAIN AND GAIN BY RADIATION AND TRANSMITTANCE FROM INSIDE SURFACE FOR TYPE *A* GLASS BLOCKS BASED ON DESIGN RADIATION CURVES AND 95 F OUTSIDE TEMPERATURE CURVE OF FIG. 4, FOR SOUTH WALL

when heat gain was most important are characteristic and show little variation. Fig. 10 shows the relation between the three sets of curves.

In making observations of solar radiation with the Eppley pyrheliometer during the past summer, the Detroit Edison Co. observed that an appreciable part of the radiation received by the Eppley pyrheliometer was scattered radiation coming from parts of the sky other than the sun or its immediate surroundings. In following up this lead the Laboratory found the same to be true and made a number of observations throughout the summer to determine the magnitude of the scattered radiation. A review of the literature indicated that Moore and Abbot of the Smithsonian Institution [3] had previously not

[3] The Brightness of the Sky, by A. F. Moore and L. H. Abbot. (*Smithsonian Miscellaneous Collections*, Vol. 71, no. 4.)

only observed the importance of scattered radiation but had made some determination of its magnitude under different conditions of location in the sky with respect to the sun, time of day and weather condition as regards haze and cloudiness.

The magnitude of scattered radiation observed at the Laboratory and by the Detroit Edison Co. during the past summer, together with a few values previously reported by the Smithsonian Institution is given in Fig. 11. The Laboratory observations were made both by shielding the disc of the Eppley
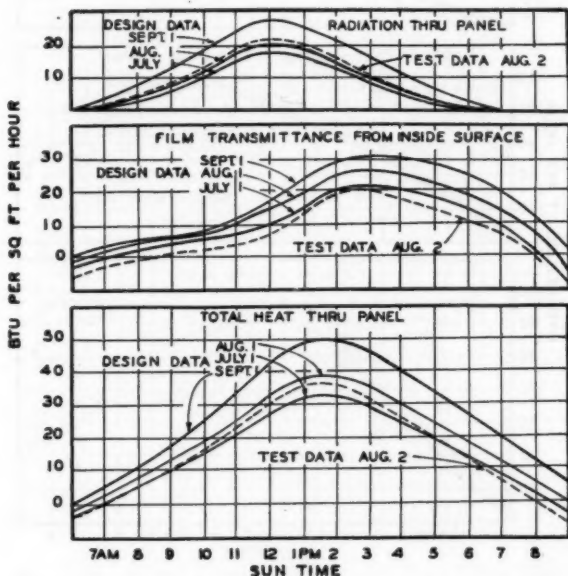
FIG. 14. JULY 1, AUG. 1 AND SEPT. 1 DESIGN DATA FOR TOTAL HEAT GAIN AND GAIN BY RADIATION AND TRANSMITTANCE FROM INSIDE SURFACE FOR TYPE C GLASS BLOCKS BASED ON DESIGN RADIATION CURVES AND 95 F OUTSIDE TEMPERATURE CURVE OF FIG. 4 FOR SOUTH WALL

pyrheliometer when mounted in a vertical position, and also by shading the disc when the instrument was mounted normal to sun radiation. Information was also had from the recorded observations at the Laboratory during the test periods when the Eppley pyrheliometer connected to a recording potentiometer received no direct radiation. Such conditions occurred when the sun was shielded by clouds, before and after the sun struck the panel, and during the afternoon of the east tests and the morning of the west test.

It will be observed that with few exceptions all of the observations made fall fairly well within the limits of the two boundary curves drawn in Fig. 11. Most of the exceptions are for observations made when the atmosphere was

hazy or when there were scattered clouds. It should be pointed out in presenting this information that this study of scattered radiation was by no means exhaustive. It is probable that a thorough study of the subject would show much greater variation for widely different conditions of haze and clouds in the sky and also some variation depending upon the angular distance of the part of the sky viewed from the sun. These variations would probably be particularly significant shortly before and after sunrise. Moreover, different results might be anticipated depending on whether the pyrheliometer was mounted horizontally so that it viewed the entire hemisphere of the sky with
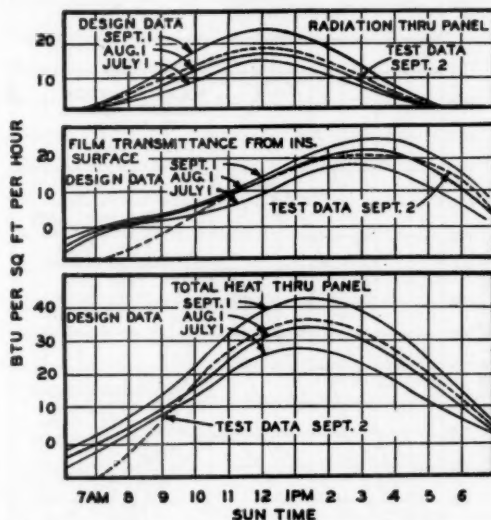
FIG. 15. JULY 1, AUG. 1 AND SEPT. 1 DESIGN DATA FOR TOTAL HEAT GAIN AND GAIN BY RADIATION AND TRANSMITTANCE FROM INSIDE SURFACE FOR TYPE *B* GLASS BLOCKS BASED ON DESIGN RADIATION CURVES AND 95 F OUTSIDE TEMPERATURE CURVE OF FIG. 4, FOR SOUTH WALL

possibly some terrestrial objects, such as buildings and hillsides along the horizon, or whether it was mounted in a vertical position facing the horizon and viewing the sky through only one quarter of the spherical angle and terrestrial surfaces through the other quarter. However, it is probable that the proposed *Curve P* for scattered radiation applying to a vertical surface will serve our purpose reasonably well for bright sunny days. The high scattered radiation observed on hazy or cloudy days should not be accepted as applying to a high solar intensity day.

During recent years there has been some difference of opinion concerning the accepted design solar radiation curve appearing in THE ASHVE GUIDE because it shows extremely high direct solar radiation. As has been pointed out on earlier occasions this curve was based upon a single day when the solar

radiation was the highest ever observed by either the Research Laboratory or the local weather bureau, whereas for design purposes it is recognized that a lower value, which might be expected at reasonably frequent intervals, would serve the purpose best. Observations at the Laboratory during the past summer revealed that on three of the brightest days the direct normal
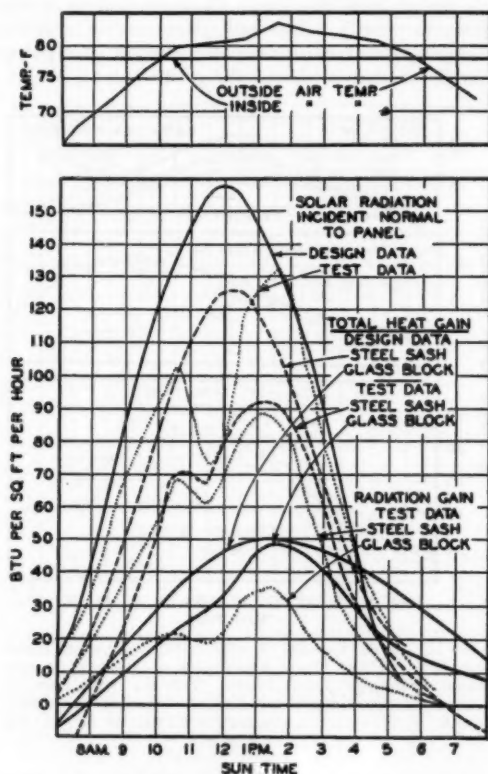
Fig. 16. Comparison Between Type *A* Glass Block and Single-Glazed Steel Sash for Both Radiation and Total Heat Gain on Aug. 25. Also Shown Are Outside and Inside Temperatures for Test Day as Well as Design Data for Total Heat Gain Through Panels

solar intensity amounted to approximately 77 per cent of the values given in The Guide. It so happens that this 23 per cent decrease just about equals the expected scattered radiation on a south wall during July and August. Hence, the maximum solar radiation intensity at noon as given by The Guide curve would appear to be about right for the total, including direct and scattered solar radiation. It should be noted that, although the direct radiation

on a vertical south wall falls off rapidly during the morning and afternoon hours and definitely ceases immediately before and after the sun appears on or leaves such walls, the scattered radiation still has considerable magnitude as long as the sun is above the horizon, and only falls off gradually as it sinks below the horizon. In this connection it is well to keep in mind that the scattered radiation referred to as coming from parts of the sky other than the sun is after all solar radiation whose path has been deflected either through reflection or refraction by clouds, moisture or other haze in the atmosphere.
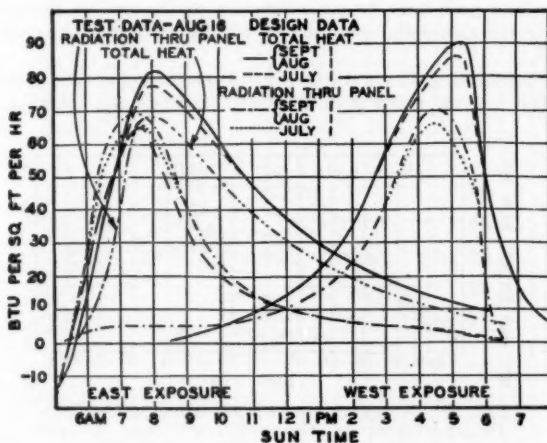
FIG. 17. TEST DATA FOR TYPE *A* GLASS BLOCKS WITH EAST EXPOSURE FOR AUG. 16 WITH DESIGN DATA FOR THESE BLOCKS FOR JULY, AUGUST AND SEPT. 1. DESIGN DATA FOR TYPE *A* BLOCK WITH WEST EXPOSURE ARE ALSO GIVEN

The entire effect of direct solar radiation and scattered radiation will therefore be referred to as solar radiation.

Suggested design solar radiation curves including both direct and scattered are shown in Fig. 12. *A* is THE GUIDE curve for direct solar radiation for a south wall on August 1. *B* is the suggested design solar radiation curve obtained by reducing the direct radiation by 23 per cent and then increasing it by the expected amount of scattered radiation as given in *Curve C* (*Curve P*, Fig. 11). *Curves D* and *E* are suggested design solar radiation curves for July 1 and September 1. These three curves are reproduced in Fig. 6 while a single point indicates the maximum on such a curve for August 12, the day for which most of the data in this figure apply.

Based upon these design radiation curves and the percentages given in Fig. 10, design curves for glass *A* are given in Fig. 13. The transmittance and total heat gain curves are also corrected for increased transmittance from the outside air to the inside air based upon the design outside air temperature curve, Fig. 6. In this figure is given also the outside air to inside air conductance for the observed day as well as that for a 95 degree-day based upon

an air to air transmittance coefficient, U, of 0.50 for a 15-mph wind. The difference between these two curves represented the amount added to the curves showing the transmittance gain from the inside surface in order to correct them for a design outside temperature cycle on a day having a 95 deg maximum. While this assumption may be questioned on the grounds that heat transfer through the panel other than by radiation has been distorted due to the heating of the glass, it is nevertheless probably true that the two effects, namely, transfer due to absorption of radiation within the glass and that due to temperature difference between outside and inside air are directly additive. At least the heat gain must be some function of the sum of these two effects. Hence, while the assumption may not be rigidly accurate, it cannot be greatly in error.

Since the total heat gain curves in Fig. 13 represent the sum of the radiation gain and film transmittance gain they also include this addition. It is of
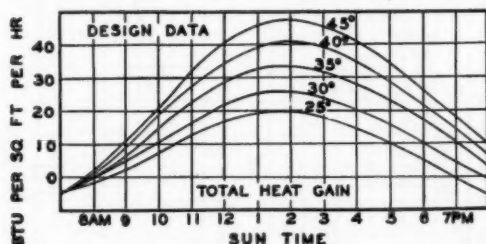
FIG. 18. DESIGN CURVES GIVING TOTAL HEAT GAIN THROUGH TYPE A GLASS BLOCK PANEL FACING SOUTH FOR AUG. 1 AT NORTH LATITUDES INDICATED

interest to note the small factor which outside to inside air transmittance is of the total gain through glass block construction when sun radiation is a factor. For convenience the actual curves based upon test data for about August 12 taken from Fig. 6 are included. Film transmittance and total gain are also shown increased by an amount assumed to represent the added gain due to the outside temperature on a normal 95 degree-day with a 15 mph wind. It will be observed that the curves, *Test data,* corrected for a 95 degree-day fall in approximately the proper location with respect to the design curves for August 1 and September 1.

In considering the suggested design curves for solar radiation intensity normal to the face of the panel, radiation gain through the blocks, and the total heat gain for a suggested design day both as regards solar radiation and outside air temperature as proposed in this paper, use is made not only of actually observed data but also of certain deductions based upon a line of reasoning. It is assumed that design solar radiation curves used in recent issues of THE GUIDE are probably high when direct solar radiation alone is considered but about right for the total radiation including scattered. The only question involved in the new design solar radiation curves suggested is, therefore, whether or not the magnitude of the values therein represents acceptable design values based upon the frequency of their occurrence. The sug-

gested design curves for radiation gain and total heat gain through the glass block depend not only upon the acceptance of the design solar radiation curves and the 95 degree-design day outside temperature curve, but also upon the assumption that the percentage curves for heat gain through the block as given in Figs. 7 to 10 apply under all conditions.

It is logical to assume that radiation through the glass block at any instant should bear some percentage relationship to the solar intensity normal to the outside surface. While this might vary somewhat throughout the day due to the angle of impingement the data collected indicate that this is not the
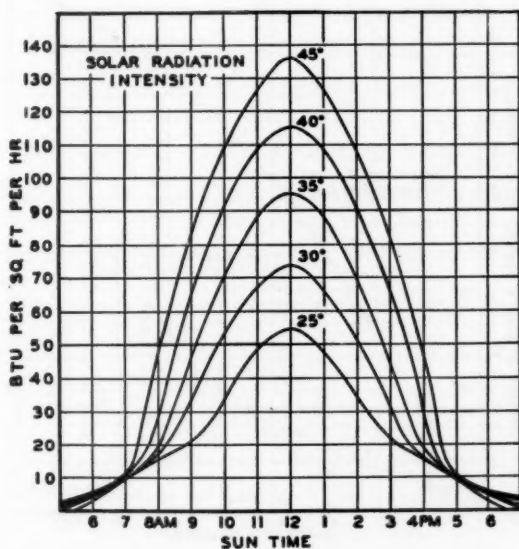
FIG. 19. DESIGN CURVES FOR SOLAR RADIATION INTENSITY IMPINGING ON SOUTH WALL FOR AUG. 1 AT NORTH LATI-TUDES OF 25, 30, 35, 40 AND 45 DEG

case; hence, the suggested design curves for radiation through the glass block may be accepted with confidence. Since the total heat gain through the glass blocks is in all cases the sum of the radiation through the block and the transmittance from the inside surface, the acceptance of the suggested design curves for the latter two values will depend upon the acceptance of the suggested percentage design curves for radiation through glass given above and the transmittance from the inside surface. Transmittance from the inside surface at any given instant is, of course, a function of the temperature differ-ence between the inside surface of the glass and the inside air. The tempera-ture of the glass blocks and therefore of their inside surface depends not only upon the current solar radiation impinging upon the outside, but also upon the inside and outside air temperature, and to a greater extent (as indicated by

the curves in Fig. 5) upon the previous condition of solar radiation impinging upon the outside or the resulting absorption of radiation within the panel itself.

As an example, if the panel were shielded from solar radiation during the day up to the time when a given observation of heat gain through the inside surface was made, this observation would be small in comparison with what it would be if the panel had not been shaded. On the other hand if the solar intensity on the outside of the panel at the instant when the observation was being made were zero, the heat gain from the inside surface would bear no relation to this intensity. However, since on any day when the solar radiation
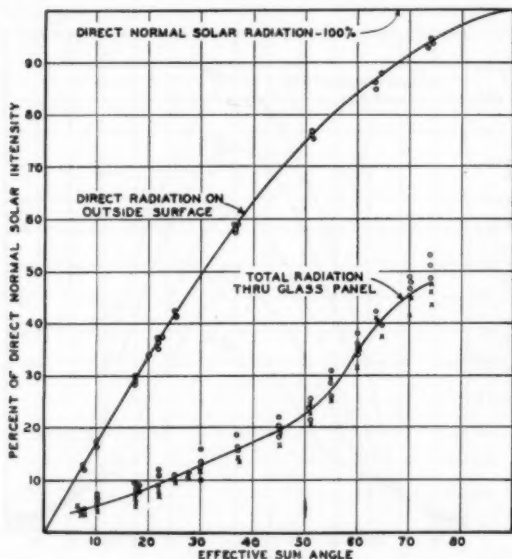
Fig. 20. Variation of Total Radiation Through Glass Panel Expressed as a Percentage of Direct Normal Solar Intensity with the Effective Sun Angle for Type *A* Glass Block. The Component of the Direct Normal Intensity Effective on the Outside Surface Is Also Shown

and air temperatures are either normal or nearly so, the history of past performance will naturally bear some definite relation to the outside solar intensity at the time. Hence, the assumption that the heat gain through the inside surface by conduction should be a certain percentage of the solar intensity on the outside appears logical and most acceptable in view of the limited information available. That this is true is borne out in part by the uniformity of the percentage curves in Figs. 7 to 10 and the close approximation between data corrected for design conditions for a given date in the summer, and actual observations when design weather conditions were approximately realized. Therefore, while these assumptions cannot be completely proven they probably hold true within narrow limits.

Calculations similar to those made for A class were carried out for the other types of block with the results shown in Figs. 14 and 15. It will be noticed that these curves all show the same characteristic shape and that they do not differ greatly from one another in magnitude. A comparison of the proposed design heat gain data for the three types of glass block studied for August 1 at the latitude of Pittsburgh is included in Table 1.

In order to determine the effect of outside shading on the total heat gain through glass blocks, a test was run on August 21 during which one of the glass block panels was completely shaded from the sun with two separated thicknesses of canvas. The total heat passing through the shaded panel between 9 A.M. and 5 P.M. amounted to 17 per cent of that passing through the
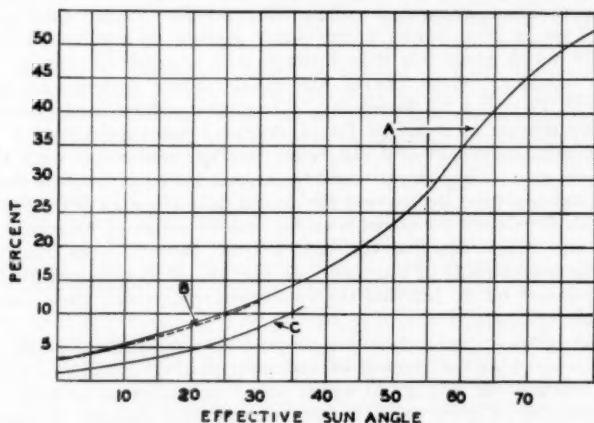
FIG. 21. PERCENTAGE OF DIRECT NORMAL SOLAR INTENSITY TRANSMITTED BY THE DIFFERENT TYPES OF GLASS AT VARIOUS EFFECTIVE SUN ANGLES

unshaded panel, while the maximum rate of heat gain through the shaded panel was 13.5 per cent of that through the unshaded one.

A test was made on August 23 during which one of the panels was shaded with an inside venetian blind having the slats set at 45 deg so as to exclude direct radiation as was done in the previous Laboratory study.[4] This reduced the total heat admitted to 70 per cent and the maximum rate of heat gain to 66 per cent of that entering through the unshaded control panel.

On August 25 a direct comparison between one of the Type A block panels and a single glazed steel sash was obtained. The results are shown in Fig. 16. It will be seen that for the actual test conditions a maximum of 94 per cent more heat entered the room containing the steel sash while for a design solar radiation and air temperature day this percentage was 147 per cent.

Two tests were made with the A glass panels facing east and one with these panels facing west. The test data for the east window, including total heat gain and radiation gain, are shown in Fig. 17 together with the design curves

[4] Loc. Cit. Note 1.

for July, August, and September 1 based on the percentage curves obtained in a manner similar to those of Fig. 10. The design curves for the total heat gain and radiation gain through a Type A glass panel facing west are also included in this figure. In the design curves shown for both the east and west tests one curve applies to July 1 while the other applies to both August and September 1. This results from the fact that the solar intensity impinging on east and west surface varies by only $2\frac{1}{2}$ per cent from August 1 to September 1. Another feature which distinguishes these curves from those pertaining to the south wall is the fact that, whereas the south wall data are applicable only to the latitude of Pittsburgh (41 deg N), the east and west data apply to all latitudes between 30 deg N and 45 deg N with an accuracy of 0.5 per cent.

The approximate manner in which the variation in the total heat gain through Type A glass blocks facing south depends on the latitude is shown in Fig. 18 which gives this gain through such a panel on August 1 at the different latitudes. These curves are based on the design solar radiation curves, Fig. 19, and a 95 degree-day. They were extrapolated from the data in the same manner as those for July 1, August 1 and September 1 (Fig. 13), the percentage curves of Fig. 10 being used in connection with the solar intensity curves of Fig. 19. It might be suggested that because of the greater range of extrapolation in deriving the design data for a 25 deg latitude from test data in Pittsburgh compared with the smaller range of extrapolation used in obtaining design data for different months, there is some question concerning the acceptability of the curves of Fig. 18. It is suggested that further work be carried out to increase the confidence with which these results may be accepted.

An interesting feature concerning the effect of the relative position of the sun with respect to the amount of radiation passing through a glass block panel is brought out in Fig. 20 which shows the variation of total radiation through a Type A glass panel in per cent of solar intensity normal to the sun as a function of the effective sun angle. The test points for the two makes of this type of glass are shown with an average curve drawn through all the points. The curve for the direct radiation falling normal to the outside surface expressed as a percentage of the solar intensity normal to the sun is also shown. Similar curves for the other types of glass block together with the curve for Type A block are shown in Fig. 21. The portion of the A curve beyond an angle of 36 deg resulted from tests with the eastern and western exposures, when the solar radiation was more nearly normal to the glass surface. Such tests were not made on the B and C types of glass.

In offering this report the authors wish to point out the obvious fact that over a period of one season it is impossible to complete all of the work which must be done before a complete understanding of all the phases of the subject can be had. Consequently, this work will be carried on over a number of years.

Danielson, H. C. Dickinson, J. E. Frazier, S. O. Hall, E. H. Hobbie, C. L. Kribs, Jr., R. A. Miller, F. W. Parkinson, J. H. Plummer, W. C. Randall, L. T. Sherwood, J. T. Staples, G. B. Watkins and F. C. Weinert.

Acknowledgment is also made of the valuable assistance rendered by the Pittsburgh Corning Corp., and the Owens-Illinois Glass Co., in making the study possible through financial support and close cooperation in carrying on the work.

DISCUSSION

H. B. NOTTAGE: I am interested in the more detailed aspects of the heat transfer characteristics of these glass blocks. When radiant energy strikes the outer surface three things will happen. Part will be reflected, part will be transmitted and part will be absorbed.

With reference to that which would be reflected, it seems that the index of refraction of the glass and the geometrical form of the outer surface would be important; and for that which would be absorbed, such physical properties as the thermal conductivity, density and specific heat of the glass would affect the observed behavior; and for that which would be transmitted, the transmission coefficient (as a function of wave length) for the radiant energy entering the glass and the nature of the inner reflecting and re-radiating surfaces would be important. It seems that ultimately we should reduce our data to some physical terms to include these factors. If such were to be brought about, the task of predicting the characteristics of other forms of blocks, blocks made of different materials, and blocks subjected to unusual conditions outside the range of test data would be greatly facilitated.

Another point of interest would be the case of heat flowing outward through the glass block. Suppose a glass block sees a cold night sky; there one would have radiation but it would be in the opposite direction. Or, suppose the blocks were covered with a coating of ice, water, or dust. Further, in some instances the blocks might receive a large amount of radiation from a selectively-reflecting surface. How would such problems be dealt with in terms of the data given?

F. C. HOUGHTEN: The reflected radiation from the outside of the glass blocks was not observed for several reasons, including the fact that there were not suitable or sufficient instruments available; neither was it possible with the limited time and funds available to establish a heat balance at the outside surface which would have been very interesting and helpful. It was possible to plot the direct radiation transmission through the blocks against the angle of impingement against the outside surface. This gives a fairly consistent curve. Naturally that radiation which was not transmitted directly was either absorbed within the block or reflected from the outside surface. No attempt was made to separate these. It is true that there are data available on the percentage radiation reflected from the outside of smooth glass surfaces based upon the angle of impingement. However, this relationship varies with the radiation wave length, and I assume that before it could be determined in any given case the exact character of the glass and the wave length of the radiant energy would have to be determined. This information would be of considerable academic interest but there is some doubt as to whether it would add much to the problem at hand, namely, the evaluation of the heat that gets into the building through the glass block and its effect on the cooling load.

In an early Laboratory study [5] some attempt was made to evaluate the heat transmitted by direct radiation through the glass in terms of that impinging on the

[5] ASHVE Research Report No. 974—Radiation of Energy Through Glass, by J. L. Blackshaw and F. C. Houghten. (ASHVE TRANSACTIONS, Vol. 40, 1934, p. 93.)

outside for different angles of incidence. While these studies were in no sense complete the percentage of the energy getting through at different angles seemed to be much more closely a function of the resulting length of the path through the glass for the different angles than of the reflection from the outside surface which our knowledge of reflection of visible light rays would indicate.

C. H. Randolph: I have two questions. First, were the glass brick that were tested the type that has the inner air space divided by this spun glass screen, which I understand is going to be used in practically all the new glass bricks?

Secondly, in our own vicinity and every place that I have been able to check them, we seem to have to use venetian blinds with the glass brick. Can you give us any further information as to what the effect of the venetian blinds were, whether they are comparable with the effect on plain glass windows?

Mr. Houghten: All six glass block tested, that is, three from each of two manufacturers, were 7¾ in. square and 4 in. thick. They represent generally all of the forms of glass block in commercial use today. These six types did not contain any glass separators or other type of fill in the air space.

One test was made by comparing a glass block panel with and without venetian blinds. The percentage reduction in the heat gain was practically the same as that found for plain glass in a previous Laboratory study.[*] It should be emphasized that the percentage gain, rather than the total amount of heat, was the same. Since the glass blocks transmitted considerably less heat, the actual heat withheld from the room containing the venetian blind was smaller but the percentage was the same.

Mr. Randolph: You mean the same percentage by that transmitted by radiation only, or by that transmitted totally?

Mr. Houghten: The test with the venetian blinds in this paper as well as in the earlier Laboratory paper gives the total reduction in heat gain from solar radiation into the cooled space.

W. C. Randall: The comparison with a single glass window is a fairly important one considering that the conclusions have been reached, based on the one test of August 25. I hope that the surface temperature of the inside of a single glass can be incorporated. I wonder if you have these data by any chance to make a comparison with the same figures on the glass block? I have a distinct impression that the surface temperature on the inside of a double glass or a glass block is higher than on the single glass window, which does not entirely check with the conclusions here, although they are probably more correct than any impression that you might get that there is such a terrific difference in the heat gain on the single window compared with the glass block.

Mr. Houghten: It is true that the temperature of the inside or outside surface of the glass block goes up much higher than the inside and outside air, whereas we showed in our studies of a few years ago that plain glass on many days rises above the outside air temperature due to the heat absorbed in the glass but by a relatively small amount. Unfortunately while there are some data given in the paper on the inside surface temperature of the glass blocks, similar data for plain glass are not available.

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[*] Loc. Cit. Note 1.

THERMAL TEST COEFFICIENTS OF ALUMINUM INSULATION FOR BUILDINGS

By GORDON B. WILKES,[*] CAMBRIDGE, MASS., F. G. HECHLER,[**] AND E. R. QUEER,[†] STATE COLLEGE, PA.

A LTHOUGH the principle of reflective insulation has been known for a long time, it was not until 1927 that thin sheets of aluminum foil were first used commercially as insulation. Because the basic principle of reflective insulation differs from that of other types and because of the relative newness of its application there is some confusion regarding its properties and use, and there are relatively few data in the literature giving the heat transmission for construction using this material.

Brightness

The term *brightness* as applied to reflective insulation ordinarily means the effect of the reflection of light on the eye. Brightness is a photometric term dealing with the measure of light (luminous power) emitted from a surface. It is synonymous with lustrous, sparkling, gleaming, flashing, glittering, or glistening. The wave lengths of the visible spectrum of light extend from approximately 0.4 to 0.7 μ (microns). The wave-length connected with heat radiation from a source at room temperature is approximately 10.0 μ, which is far removed from the visible range. If there were a definite relation between the reflectivity of light and this long wave-length radiation, then the reflectivity for infrared could be judged by means of the eye and visible reflectivity. Unfortunately, there is no such definite relation. It has been stated [1] that "a piece of aluminum may have high reflectivity for visible light and low reflectivity for infrared radiation, or it may have only fair reflectivity for light and be an excellent reflector of infrared radiation, depending on the presence or absence of surface films. It may also be a good reflector for both kinds of radiation."

A mirror, consisting of glass with a *silvered* surface on the back of the

outside for different angles of incidence. While these studies were in no sense complete the percentage of the energy getting through at different angles seemed to be much more closely a function of the resulting length of the path through the glass for the different angles than of the reflection from the outside surface which our knowledge of reflection of visible light rays would indicate.

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A mirror, consisting of glass with a *silvered* surface on the back of the

glass, is an excellent reflector of light but it is a very poor reflector of infra-red radiation corresponding to room temperature. In fact, such a mirror would have about the same reflectivity for infrared as a heavy coating of black paint.

With this in view, it is obviously impossible to judge the infrared reflectivity or emissivity of a surface by its appearance to the eye. Consequently, in a discussion of reflective surfaces for building insulation, the term *brightness* has no specific meaning. The terms *emissivity* or *reflectivity* definitely define the radiating or reflecting power of a surface and values may be determined directly for the long wave-length radiation corresponding to room temperatures.

Permanence

Since the insulating value of reflective surfaces is due largely to the nature of the surface, it is only natural that inquiry should be made as to the permanence of these surfaces. The following quotation from a previous paper [2] will show some of the reasons why the life of aluminum foil insulation need not be doubted when properly installed under normal conditions.

Aluminum foil has now been in use for a sufficient number of years to establish the permanence of the surface when properly installed under suitable conditions. The following examples are taken from sources that are believed to be authentic:

1. Aluminum-foil-insulated house in Skovshoved, Denmark. This house was built and insulated in 1927. Samples of the foil were removed in October, 1935, by engineers from the Tecnolgisk Institute, Copenhagen. The official report states: "We can testify that the brightness of the aluminum foil was unchanged in comparison to new foil of the same kind."

2. Aluminum-foil-insulated provision chambers of the motorship *Leverkusen,* fitted out in May, 1928. In January, 1934, samples of this foil were removed by representatives of Lloyd's Register of Shipping, Hamburg; they reported: "The general impression gained as a result of the examination made is that the insulation examined by us is in exactly the same condition as it was five years ago when built into the vessel."

3. Aluminum-foil-covered cardboard taken from foil-insulated dry-ice cabinet. It was exposed to the weather on a beach at Coney Island from September, 1934, to April, 1935. The emissivity of the foil removed from this cabinet was 0.04.[3]

4. Aluminum foil suspended vertically in laboratory for three years and measured with accumulated dust and fume. The emissivity was 0.05.[4]

The following examples give some of the author's personal experience over a period of 10 years with regard to the permanence of aluminum foil insulation:

1. Aluminum-foil-insulated residence, Wellesley Hills, Mass., built and insulated in 1933. The author personally removed samples of this insulation from the underside of the roof in June, 1938. The samples appeared in perfect condition, and the average emissivity of the four samples removed was 0.054 as compared with 0.045 for new foil.

2. Aluminum foil after two-year exposure to salt spray and moisture on underside of roof of log boat house in Newington, N. H. The foil was spotted with salt that had been left on the foil by evaporation of the salt spray. The emissivity was found to be 0.10.

3. Aluminum foil exposed in a vertical position since 1929 to the dust and fumes in the Heat Measurement Laboratory, M.I.T. Samples of this foil have been removed from time to time and the emissivity has been determined. Over a period of 10 years no appreciable change in emissivity has been found.

4. Aluminum foil insulation placed over ceiling in attic of residence in Newton Centre, Mass. After three years no appreciable change could be noted.

[2] Reflective Insulation, by G. B. Wilkes. (*Industrial and Engineering Chemistry*, 31: 832, July, 1939.)
[3], [4] Loc. Cit. Note 1.

GUARDED PLATE CALORIMETER
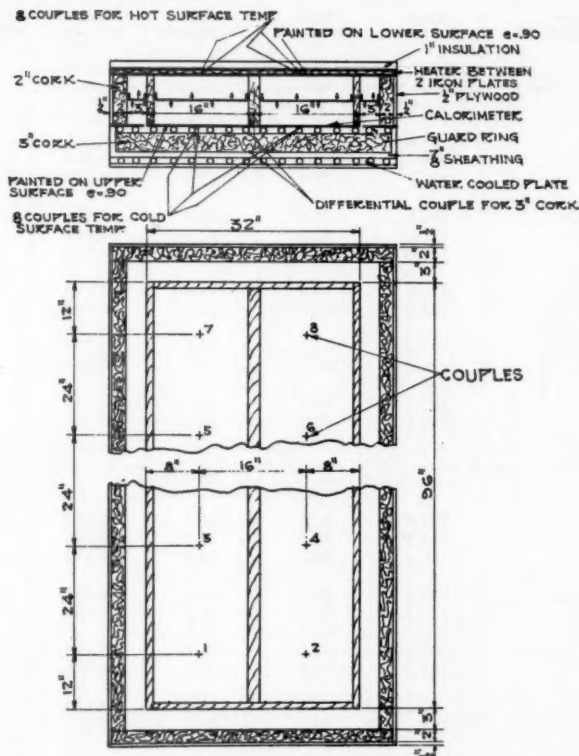
FIG. 1. DIAGRAM OF GUARDED PLATE CALORIMETER

5. Aluminum foil insulation in ceiling of a cellar with no covering of any kind. After three-year exposure this foil appeared in perfect condition except over the laundry tubs where soap had apparently come in contact with the foil and attacked the surface in spots. This cellar was very damp every summer.

6. Hundreds of samples of aluminum foil have been stored in the laboratory for various periods of time up to 10 years with no visible signs of deterioration of the surface.

There are, however, certain conditions, that may seriously affect the heat resisting properties of aluminum foil. Alkalies attack aluminum readily and the foil should always be protected from direct contact with wet plaster. If prolonged exposure to moisture is expected, it is possible to coat aluminum foil with a thin transparent lacquer as a protective agent. This coating,

TABLE 1. RESULTS OF CALORIMETER TESTS

POSITION	WARM SURFACE F	COLD SURFACE F	TEMP. DIFFERENCE F	CON- DUCTANCE (TEST)	CAL- CULATED U^a
TYPE II INSULATION MAKING TWO 4-IN. AIR SPACES. INSTALLED AS IN CONSTRUCTION NOS. 1, 2, AND 3 (FIG. 3)					
Horizontal.........	98.6	82.4	16.2	0.081	0.066
Heat Down.........	107.0	80.2	26.8	0.089	0.070
	125.2	79.8	45.4	0.097	0.075
30 Deg Slope........	107.5	79.5	28.0	0.135	0.105
Heat Down.........	127.6	80.5	47.1	0.149	0.114
	132.0	78.0	54.0	0.156	0.118
Vertical...........	110.6	79.7	30.9	0.196	0.124
	125.0	82.0	43.0	0.207	0.129
30 Deg Slope........	109.2	81.5	27.7	0.229	0.156
Heat Up...........	129.3	82.3	47.0	0.248	0.165
Horizontal.........	105.7	80.1	25.6	0.238	0.140
Heat Up...........	126.1	82.7	43.4	0.266	0.149
TYPES I AND II INSULATION MAKING THREE 2.7-IN. AIR SPACES. INSTALLED AS IN CONSTRUCTION NOS. 4, 5, AND 6 (FIG. 3)					
Horizontal.........	104.8	82.1	22.7	0.063	0.053
Heat Down........	115.1	80.0	35.1	0.067	0.056
	145.3	80.5	64.8	0.072	0.059
30 Deg Slope........	106.6	80.3	26.3	0.088	0.075
Heat Down.........	128.2	82.0	46.2	0.097	0.081
Vertical...........	100.7	77.7	23.0	0.114	0.086
	121.7	76.0	45.7	0.134	0.096
30 Deg Slope........	99.1	77.0	22.1	0.133	0.105
Heat Up...........	118.2	78.1	40.1	0.159	0.120
Horizontal.........	105.3	78.1	27.2	0.158	0.108
Heat Up...........	130.8	83.1	47.7	0.176	0.116

a The calculated U value is based on the following values taken from THE GUIDE, 1939:
Horizontal—flat roof, page 112, No. 6, 1½-in. roof deck, metal lath and plaster—uninsulated—$U = 0.26$
30 deg-sloping roof, page 114, No. 6, asphalt shingles, metal lath and plaster—uninsulated — $U = 0.34$
Vertical-frame wall, page 107, No. 65(B), metal lath and plaster—uninsulated — $U = 0.26$
Ordinary air space — $C = 1.10$

however, should only be applied by the manufacturer as an ordinary coat of transparent lacquer will usually ruin the reflective properties of the aluminum for infrared radiation although it will still appear bright to the eye and will continue to be an effective reflector of visible radiation.

A heavy coating of dust on a surface of aluminum will certainly reduce the reflectivity materially. In ordinary building insulation, this is not likely to occur on vertical walls nor on the underside of sloping and horizontal sheets. On the upper side of sloping or horizontal surfaces it would be possible for a heavy dust layer to collect if there were sufficient dust available, which is

ROOF & WALL COEFFICIENTS WITH ALUMINUM INSULATION

TYPE OF CONSTRUCTION	CONST. No	HEAT FLOW	U	C₁	TEMPERATURE ΔT FOR C₁
$\frac{7}{8}$" WOOD, 2"x 6" RAFTERS, WIRE LATH & PLASTER, ROOFING, ROOF SHEATHING	1	UP	0.14	0.24	25
		UP	0.15	0.27	45
		DOWN	0.07	0.08	25
		DOWN	0.08	0.10	45
2"x 6" RAFTERS, WIRE LATH & PLASTER, ROOFING, ROOF SHEATHING, 30°	2	UP	0.16	0.25	25
		UP	0.16	0.25	45
		DOWN	0.10	0.15	25
		DOWN	0.11	0.15	45
BUILDING PAPER, 8" PINE LAP SIDING $\frac{1}{2}$" BUTT, $\frac{7}{8}$" FIR SHEATHING, 2"x 8" STUDDING, $\frac{7}{8}$" PLASTER BASE	3		0.12	0.19	25
			0.13	0.21	45
$\frac{7}{8}$" WOOD, 2"x 6" RAFTERS, WIRE LATH & PLASTER, ROOFING, ROOF SHEATHING	4	UP	0.11	0.16	25
		UP	0.12	0.17	45
		DOWN	0.05	0.06	25
		DOWN	0.06	0.07	45
2"x 6" RAFTERS, WIRE LATH & PLASTER, ROOFING, 30°	5	UP	0.11	0.14	25
		UP	0.12	0.17	45
		DOWN	0.07	0.09	25
		DOWN	0.08	0.10	45
BUILDING PAPER, 8" PINE LAP SIDING $\frac{1}{2}$" BUTT, $\frac{7}{8}$" FIR SHEATHING, 2"x 6" STUDDING, $\frac{7}{8}$" PLASTER BASE	6		0.09	0.12	25
			0.10	0.13	45

Fig. 2. Results of Calorimeter Tests

Fig. 3. (Right) Roof and Wall Coefficients with Aluminum Insulation

CONDUCTANCE—B.T.U., HR⁻¹, FT⁻², °F⁻¹

TEMPERATURE DIFFERENCE °F ACROSS 8" SPACE

8 INCH AIR SPACE ONE LAYER TYPE I MIDWAY IN SPACE

8" ONE LAYER TYPE II

EQUALLY SPACED

unlikely in most cases. If a sheet of building paper is placed above the top layer of foil in these sloping and horizontal positions, even this possibility would be minimized. A light coating of dust, while affecting the visible reflectivity considerably, has only a minor effect on the infrared reflectivity.

Heat Transmission Tests on Roof and Wall Sections with Aluminum Insulation

The insulation used in these tests was of two kinds. One consisted of a sheet of kraft paper with aluminum foil cemented to one side only (Type I) while the other had aluminum foil on both sides of the paper (Type II). The material was purchased in the open market and was representative of that now available to the consumer. Prior to making the heat-transfer tests, samples were taken from each roll of material and emissivity measurements were made with a radiometer. The values ranged from 0.045 to 0.053 with an average value of 0.05 for a temperature of 212 F.

Roof Tests [5]

The essential parts of the apparatus used in the calorimeter tests have been previously described.[6] Important improvements were made, however, for the purpose of the tests which include the following:

1. A 5-in. water-cooled guard ring was installed entirely around the calorimeter.
2. A water-cooled plate was placed on the back, separated from the calorimeter by 3 in. of corkboard. This minimized the gain or loss of heat from the calorimeter and made this correction very small.
3. The air space in the guard ring was of the same construction as in the test area.
4. An electrically-heated plate, covering the calorimeter and guard ring, was installed on the warmer side of the air space as shown in Fig. 1.

The results of these calorimeter tests (Table 1) are shown best in Fig 2. Each point indicates an actual test value which is the average of two or more runs which checked better than 1 per cent. It required approximately 24 hours to establish thermal equilibrium after the position or temperature condition was changed.

The calorimeter tests, as shown in Fig. 2, confirm previous evidence that the rate of heat transfer across air spaces faced with reflective surfaces is greatly affected by the position and direction of heat flow. In the case of a flat roof, the conductance of an 8-in. air space with a single layer of reflective insulation placed in the middle (temperature difference being 25 F) is 0.087 Btu per hour per square foot per degree Fahrenheit when the heat flow is downward as compared with 0.238 with the heat flow upward. In other words, under winter conditions the conductance is equivalent to about 1.2 in. of conventional insulation but under summer conditions it is equivalent to more than 3 in. of conventional insulation.

Fig. 2 also shows the variation of the conductance with temperature difference, indicating in general a very definite increase with increase of temperature difference.

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[5] The emissivity and roof tests were made at Massachusetts Institute of Technology under the supervision of Professor Wilkes.
[6] Radiation and Convection Across Spaces in Frame Construction, G. B. Wilkes, and C. M. F. Peterson. (ASHVE Transactions, Vol. 43, 1937, p. 351.)

This type of apparatus is capable of considerable precision as may be seen from the data and sample calculation. All of the factors used in calculating the conductance value may be measured with sufficient precision so that the error should certainly be less than 1 per cent. Separate runs can easily be made to check within 1 per cent and usually within $\frac{1}{2}$ of 1 per cent. Of course, this merely means that results with this type of apparatus can be duplicated. Every precaution has been taken to insure the absolute value of the conductance that could reasonably be installed.

Data and Calculation of Sample Run. Horizontal—heat downward, Types I and II in 8-in. space.

Couple No.[a]	Calorimeter Temp. MV	F	Heater Temp. MV	F	Calorimeter Time	Δt MV
1	1.108	80.05	1.932	113.90	0	0.102
2	1.102	79.78	1.929	113.79	1	0.102
3	1.103	79.82	1.968	115.34	2	0.102
4	1.099	79.64	1.979	115.79	3	0.102
5	1.102	79.78	1.991	116.28	4	0.102
6	1.104	79.87	2.015	117.25	5	0.102
7	1.108	80.05	1.936	114.06	6	0.103
8	1.105	79.91	1.957	114.89	7	0.102
...	...	...	...	...	8	0.102
...	...	...	...	...	9	0.103
...	...	...	...	...	10	0.102
Avg.....	...	79.86	...	115.16	..	0.1022

[a] For location see Fig. 1.

Rise in temperature of calorimeter water 0.1022 mv (five couples in
  series).. 0.901 F
Weight of calorimeter water................................. 55.86 lb per hour
Uncorrected heat absorbed by calorimeter.................... 50.2 Btu per hour
Temperature difference across 3-in. cork back.............. 0.13 F
Heat loss from calorimeter through cork back............... 0.3 Btu per hour
Corrected heat absorbed by calorimeter..................... 50.5 Btu per hour
Temperature difference across 8-in. space.................. 35.30 F
Area of calorimeter.. 21.3 sq ft

Conductance of space $\dfrac{50.5}{21.3 \times 35.30} = 0.0672$ Btu per hour per square foot per degree Fahrenheit

Fig. 3 shows the results reduced to a form in which they are usually used. The U values shown are for an inside surface coefficient of 1.65 and an outside coefficient of 6.00 corresponding to a 15 mph wind.

Wall Tests [7]

The tests on vertical wall sections using 2 x 4 studs were made in a large guarded hot-box apparatus designed for sections $67\frac{5}{8}$ in. x 67 in. in area. A sketch of the equipment with a wall in position ready for test is shown in

[7] The wall tests were made by the Engineering Experiment Station, Pennsylvania State College, under the supervision of Professors F. G. Hechler and E. R. Queer.

Fig. 4. The test box covered a 4 ft x 4 ft area at the center of the wall under test. The following features may be noted:

1. The heat to the boxes was supplied by 440-volt strip heaters located at the bottom of each box and shielded. The heaters were operated on low voltage a-c to keep radiant heat at a minimum.

2. Sensitive bimetallic thermostats controlled a small portion of the total heat input to maintain the temperatures constant within ± 0.2 F.

3. The inner and outer surfaces of the test box were covered with 16-oz copper sheets, painted black, thermally and electrically insulated from each other. Thermo-
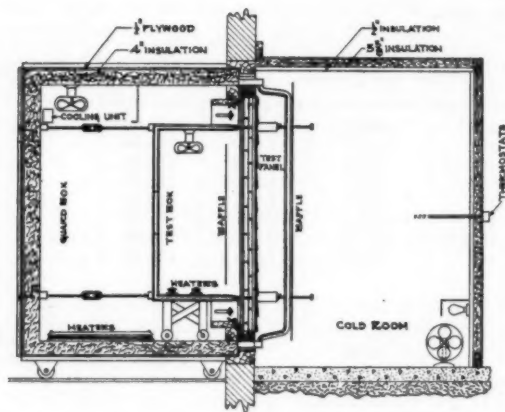
FIG. 4.   GUARDED HOT BOX ASSEMBLY WITH TEST PANEL
IN POSITION

couples were soldered to both surfaces and used for balancing the temperatures in the test box and guard box.

4. The power input to the test box was measured with a portable standard watt-hour meter.

5. To permit testing with the warm side of the panel at temperatures below normal room temperature a small water-cooled automobile heater was installed near the top of the guard box.

6. Access doors were provided at the back of the guard box and the test box to permit inspection and adjustment when installing a test panel.

7. Small circulating fans were installed in the test box, the guard box, and the cold room. It was found that those in the test box and guard box were not needed to give satisfactory temperature distribution and they were not operated. The fan in the cold room was run at low speed to maintain a slow air movement.

8. Baffles or shields were placed near the test wall on each side.

Fig. 5 shows the construction of the test wall and the method of taking it apart for the installation of the insulation in different ways. The 4 ft x 4 ft test box covered three stud spaces on the panel. To limit the vertical height in the stud spaces to 4 ft, in order to separate the test and guard areas, thin strips of plywood were used to block off these spaces, as shown. Thermocouples were attached to both the hot and cold sides; other thermocouples

were distributed throughout the wall, making it possible to determine the conductances of all parts of the wall.

The results of the guarded hot-box tests are shown in Table 2. Air temperatures in the tests at 0 and 70 were used to simulate those encountered in

WALL TEST PANEL
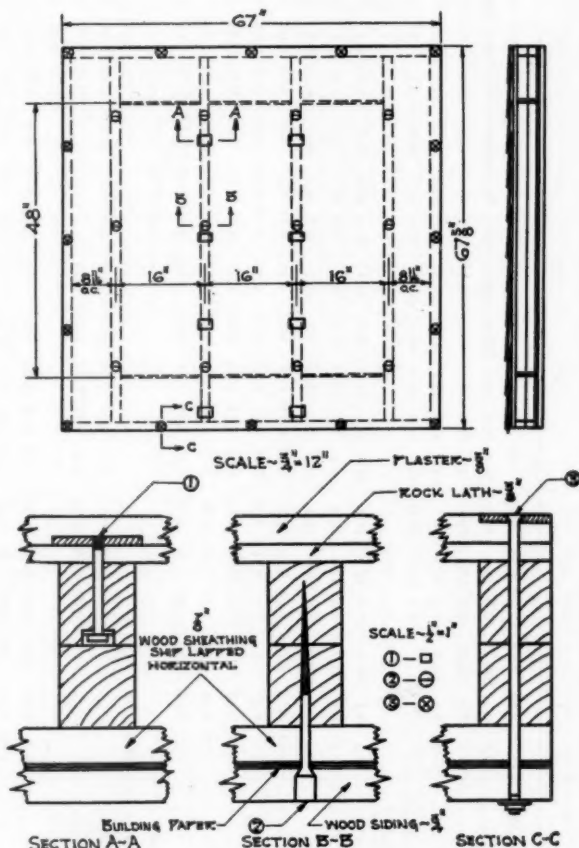
Fig. 5. Construction of Wall Test Panel

.heating plant design; 30 and 70 are an average winter condition; 70 and 100 are a maximum summer condition. It required about 14 hours to establish thermal equilibrium for each test and equilibrium was maintained for 10 hours.

Fig. 6 shows the results reduced to the form in which they are most fre-

TABLE 2. RESULTS OF THE GUARDED HOT-BOX TESTS

	Wall No. 1			Wall No. 2			Wall No. 2ᵃ					Wall No. 3			Wall No. 4			Wall No. 5ᵃ		
Heat, Btu per sq ft per hour	13.25	7.64	6.15	7.99	4.45	3.21	7.66	7.74	4.21	3.26	8.95	9.47	5.44	4.41	5.85	3.19	2.37	9.20	5.20	4.06
Hot Air, F	69.9	70.3	100.3	70.0	70.4	100.3	70.2	70.3	70.3	100.4	140.0	70.1	70.3	100.3	70.1	70.3	100.1	69.7	69.9	100.1
Hot surface, F	59.3	63.9	95.5	64.1	67.3	98.0	64.4	64.4	67.2	98.2	135.1	63.5	66.1	97.1	65.9	67.9	98.4	62.8	65.8	97.2
Plaster base, F	53.6	60.4	92.4	61.1	65.7	96.7	61.4	61.6	65.7	97.0	131.9	59.7	63.8	95.2	63.8	66.6	97.3	58.9	63.7	95.4
(1) Aluminum surface, F							45.2	44.4	54.6	88.2	113.0	55.4	61.5	93.9	49.5	57.8	90.7	50.8	59.1	92.3
(2) Aluminum surface, F				46.4	54.7	88.0									35.2	45.9	83.7			
Sheathing, F	41.1	53.1	86.9	25.9	42.9	78.8	25.8	24.8	43.0	79.2	93.1	31.4	46.8	81.6	19.3	39.2	76.3	28.1	44.8	80.4
Cold surface, F	12.3	36.5	74.2	8.4	33.6	72.2	8.4	7.2	33.7	72.6	75.9	9.5	34.7	72.2	6.3	32.5	71.5	7.8	34.0	72.6
Cold air, F	0.0	30.1	70.3	0.0	29.9	70.1	+0.2	−0.6	30.3	70.5	70.5	−0.6	29.7	69.4	−0.3	30.0	70.0	−0.8	30.0	70.2
U at 15 mph. (calculated)	0.23	0.23	0.24	0.13	0.12	0.11	0.12	0.12	0.12	0.12	0.14	0.15	0.15	0.16	0.09	0.08	0.08	0.15	0.15	0.15
U (test)	0.19	0.19	0.21	0.11	0.11	0.11	0.11	0.11	0.11	0.11	0.13	0.13	0.13	0.14	0.08	0.08	0.09	0.13	0.13	0.14
C, surface to surface	0.28	0.28	0.29	0.14	0.13	0.12	0.14	0.14	0.13	0.13	0.15	0.18	0.17	0.18	0.10	0.09	0.09	0.17	0.16	0.17
f, inside	1.25	1.19	1.28	1.35	1.44	1.40	1.32	1.31	1.36	1.48	1.83	1.43	1.30	1.38	1.39	1.33	1.39	1.33	1.27	1.40
C, plaster and base	2.32	2.18	1.98	2.66	2.78	2.47	2.55	2.76	2.81	2.72	2.80	2.49	2.37	2.32	2.79	2.45	2.16	2.36	2.48	2.26
C, plaster base to Type I												2.20	2.37	3.39						
C, air space, e-eff = 0.82	1.06	1.05	1.12															1.14	1.13	1.31
C, (1) aluminum air space				0.54	0.40	0.37	0.47	0.45	0.38	0.37	0.47	0.40	0.37	0.36	0.41	0.36	0.36	0.41	0.36	0.34
C, (2) aluminum air space				0.39	0.38	0.35	0.40	0.40	0.36	0.36	0.45				0.41	0.36	0.34			
C, (3) aluminum air space															0.37	0.33	0.32			
C, lumber	0.46	0.46	0.48	0.46	0.48	0.49	0.44	0.44	0.45	0.49	0.52	0.43	0.45	0.47	0.45	0.48	0.49	0.45	0.48	0.52
f, outside	1.08	1.19	1.58	0.95	1.20	1.53	0.93	0.99	1.24	1.55	1.66	0.94	1.09	1.58	0.89	1.28	1.58	1.07	1.30	1.69
C, plaster base to sheathing	1.06	1.05	1.12	0.23	0.20	0.18	0.22	0.21	0.19	0.18	0.23	0.34	0.32	0.33	0.13	0.12	0.11	0.30	0.28	0.27

ᵃ Wall No. 5—same insulation as Wall No. 3, except that it was looped between the studs, forming a 1-in. air space.
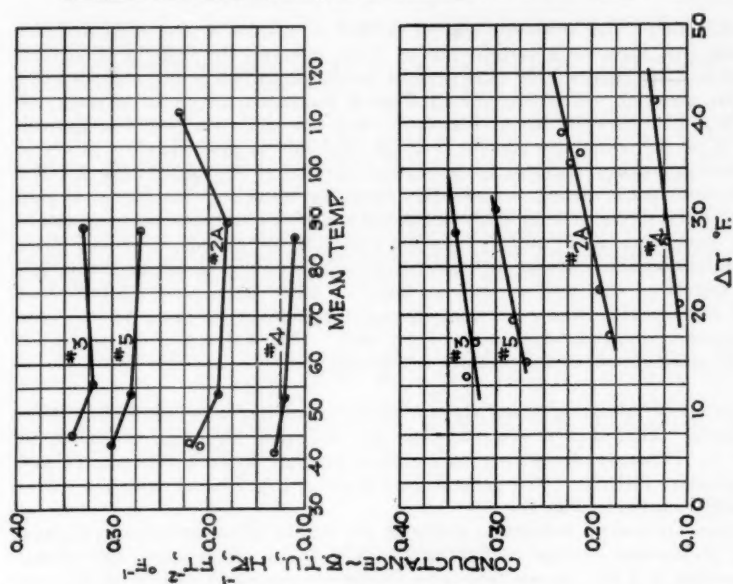
Fig. 7. RELATION BETWEEN CONDUCTANCE AND TEMPERATURE DIFFERENCE

WALL COEFFICIENTS WITH ALUMINUM INSULATION

TYPE OF WALL	WALL N°	C	U	C_1	AIR TEMPERATURES
BUILDING PAPER, 8" PINE LAP SIDING 1 BUTT, 13/16" FIR SHEATHING 2"×4" STUDDING, 1/2" PLASTER BASE	1	0.28	0.25	1.06	0 / 70
		0.28	0.25	1.05	30 / 70
		0.28	0.24	1.12	100 / 70
BUILDING PAPER, 8" PINE LAP SIDING 1 BUTT, 13/16" FIR SHEATHING 2"×4" STUDDING, 3/8" PLASTER BASE	2a	0.14	0.12	0.21	0 / 70
		0.15	0.12	0.19	30 / 70
		0.15	0.12	0.18	100 / 70
BUILDING PAPER, 8" PINE LAP SIDING 1 BUTT, 13/16" FIR SHEATHING 2"×4" STUDDING, INSULATION, 3/8" PLASTER BASE	3	0.18	0.15	0.34	0 / 70
		0.17	0.15	0.32	30 / 70
		0.18	0.16	0.35	100 / 70
BUILDING PAPER, 8" PINE LAP SIDING 1 BUTT, 13/16" FIR SHEATHING 2"×4" STUDDING, 3/8" PLASTER BASE	4	0.10	0.09	0.13	0 / 70
		0.09	0.08	0.12	30 / 70
		0.09	0.08	0.11	100 / 70

Fig. 6. WALL COEFFICIENTS WITH ALUMINUM INSULATION

quently used. The insulation applied to Wall No. 2 had a row of ⅛-in. holes in one stud space on a vertical line at 12-in. intervals through the insulation. Wall No. 2a contained the same type of insulation as No. 2, but without holes in the material. Wall No. 5 had Type I insulation looped in between the studs similar to Wall No. 2, Fig. 6. A test was made on Wall No. 2a with 140 F on the warm side and 70 F on the cold side to show that for reflective insulation nothing is gained by testing the material at high temperatures. The tests were made during the summer when the humidity was high. Before making the second test on Wall No. 2a (0 and 70 F) the test-box temperature was dropped to 50 F with 0 F on the cold side for two days. Under these conditions moisture might be expected to collect on the insulation; however, the results of this test, as well as those of other tests, gave no indication that condensation was occurring. Some variations took place in the conductance of the individual parts of the wall. This was caused by variations in the small measured temperature differences.

For most insulating materials the conductance increases with mean temperature. That this is not the case for reflective insulation is shown in Fig. 7. However, there is a well defined relation between conductance and temperature difference. In general the curves in Fig. 7 agree with the corresponding curves in Fig. 2.

There are several interesting points in the data to which attention is invited. The plaster-side surface coefficient for a 70 F air temperature for natural convection is 1.33. At one time The Guide recommended a value of 1.34 but the present value is 1.65. This difference is practically insignificant for the average wall. An average conductance of 2.54 was obtained for the plaster and base for an inside air temperature of 70 F. In Wall No. 3 the insulation was not attached directly to the plaster base, but was stretched tightly across the studs which produced a certain amount of resistance to heat flow between the plaster base and paper. This resistance was included in the value for C of Wall No. 3, Fig. 6. The resistance of about 0.43 obtained in these tests agrees closely with that given in a previously published paper.[8]

Summary

1. When the roof tests were made in the various positions, one test of each application of insulation was made in a vertical position. An inspection of Figs. 3 and 6 shows the excellent agreement obtained in the results by the two different methods of test. These values confirm the reliability of the guard plate calorimeter as a method of making heat transmission tests and establish the validity of the results obtained by two independent methods.

2. It should be noted that the results of these tests again confirm the fact that the conductance of an air space faced with a reflective surface varies with the temperature difference and, as shown in these experiments, there appears to be no relation with mean temperature. In the case of reflective insulation the principal mode of heat transfer is by convection and only a small amount is transferred by radiation. The heat transfer by convection varies as the 5/4 power of the temperature difference. Consequently the smaller the temperature difference the lower the conductance value of an air space containing reflective insulation. This fact is borne out by the data and in all cases the values at 25 F difference are lower than those for 45 F difference.

[8] Heat Loss through Wall Construction, by Alf Kolfaath. (ASHVE Journal, September, 1924, p. 627.)

The 25 F difference is an extreme summer condition and the 45 F difference is an extreme winter condition.

3. For reflective insulation orientation must be considered. An unusually low heat-transfer value is obtained with the heat flowing down. This is particularly striking in the horizontal positions, Constructions Nos. 1 and 4, Fig. 3. In these cases the major portion of the heat transfer is by conduction while radiation and convection are a minimum. In some instances the reflective insulated structures are vented for summer conditions. However, the vents should be closed for the heating season.

4. Insulation is usually added to a structure to obtain fuel savings and produce comfort to the occupants of the building. The comfort factor is more important in the summer than winter, since most heating systems have sufficient capacity to produce comfort. Since reflective insulation performs most effectively with the heat flowing down, as it does in the summer, this insulation is particularly effective in retarding summer heat influx through roofs into top floor living quarters.

ACKNOWLEDGMENT

The authors wish to acknowledge the assistance given by Prof. C. M. F. Peterson and L. R. Vianey of the Heat Measurements Laboratory, Massachusetts Institute of Technology; and W. H. Ness, research assistant in engineering research, Engineering Experiment Station, The Pennsylvania State College, in making these tests.

DISCUSSION

P. F. McDERMOTT [a] (WRITTEN): Although the principle of reflective insulation has been known for a long time it had little practical appeal until thin metallic foils, notably aluminum, were available at low cost. More or less immediately thereafter, patents were taken out, papers written, and applications attempted. A revolution in insulation practice was thought likely by numerous proponents of reflective insulations and quite a rush resulted to be able to supply a material of the reflective type. In an effort to get in with this trend, even a material labeled non-metallic reflective insulation was brought into the market.

It would be interesting to see the service record of all these attempts to make use of foil, and to establish a relation between the quantity used in these experiments and that used as regular permanent insulation as a result of these experiments. Unfortunately, such information has never been made available. Thus, we talk of foils as *when new* or in terms of *laboratory tests* into which latter classification the present paper falls practically by title.

The authors cite ten cases in which the foil apparently has stood up for from six months to 10 years under various conditions.

It is significant that one sample after six months at Coney Island had an emissivity of 4 per cent and another which after a two-year exposure on the underside of a boat house roof had an emissivity of 10 per cent. These results differ materially from weathering tests made in New York, Bayonne, and Seabright, N. J., where the exposed foil was practically destroyed in six months. In general we believe that for most examples, where foil has succeeded in withstanding service conditions, examples of failure can be cited. However, were the failures only a few per cent of the trials, the foil would not be acceptable when competing with other types of insulations.

Permanence of aluminum for cooking ware and other household products is well known, the resistance to corrosion being attributed to the formation of a protective

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oxide coating. However, for the thin foils, offered for insulation, corrosion results in pinholing and failure before a protective oxide film is established. Lacquered foils have been made so as to prevent this failure but they have somewhat higher emissivities and will fail due to the lack of complete coverage of the lacquer and to the failure of the lacquer itself.

Visual examination of reflective insulations means little as pointed out in a recent paper.[10] This requires that highly specialized apparatus be available to check on reflective type material, whereas the consumer can by visual and manual means detect relatively small variations in the quality of the more common types of insulation.

The efficiency of a foil installation may vary in other ways also, as for example foil to result in an emissivity of 25 per cent.

The direction of heat flow is another interesting factor. Recently a home owner requested us to recommend insulation to supplement the foil in the walls and flat roof of his house. The house was cool in summer but cold walls and ceilings in winter caused discomfort. There was no access for inspection of the condition of the foil. Assuming the foil to be in good condition, it was concluded that the roof was well insulated against downward heat flow which resulted in a cool house in summer. With heat flow upward as in winter, however, the roof treatment was not efficient. Furthermore, the inadequate wall insulation which had had little influence on summer results, seriously affected the winter results. Treatments that involve directions of heat flow will frequently require special attention to assure proper results.

R. D. WATT: It is presumed that the foil which was tested and from which these coefficients were derived was carefully installed, perhaps not up to a laboratory standard but certainly representative of good practice. I should like Professor Queer's opinion as to how much variation there would be in these coefficients from what we might call an average or a hurry-up job of installing the foil.

C.-E. A. WINSLOW: I would like to know whether the $\frac{5}{4}$ factor in the formula was put in to account for heat loss through the apparatus. Newton's law of cooling ought to be a straight line relationship. I am wondering what that $\frac{5}{4}$ meant.

G. B. WILKES: The question about the difference between laboratory and practical application is an old one. In experiments at Massachusetts Institute of Technology, and at Penn State, there is no question but what the insulation was installed in first-class shape. Nothing was sealed, but the insulation was attached in normal fashion as you would make any laboratory test. In practice, if the work is rushed and foil was left hanging loose, there is no question but what there would be a difference in the results. For example, in Cambridge, Mass., a builder mounted aluminum foil on plasterboard facing the inside of the room and put wallpaper over it. Obviously there would be no advantage from that type of foil insulation.

The answer to the question about Newton's law of cooling, is that when you are dealing with an air space faced with a reflective material, the heat transfer is primarily by convection except when there is downward flow—then there may be considerable conduction.

Convection does not follow Newton's law, but as far as we know at present based purely on experimental evidence it increases approximately with the $\frac{5}{4}$ power of the temperature difference. Consequently, the conductance of an air space lined with foil is not proportional to the temperature difference, but it does increase with the $\frac{5}{4}$ power of the temperature difference.

With low temperature differences you have low values and at high temperature differences you have higher values and, as Professor Queer pointed out, it is the

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[10] Loc. Cit. Note 1.

temperature difference and not the mean temperature that affects the conductance value over the range covered by these experiments.

H. B. NOTTAGE: Would the value of these over-all coefficients be enhanced if they could be broken down exactly into the component mechanisms of convection, radiation and conduction? I might suggest a method which could be used to accomplish that. It is one originally developed by Nusselt, in which a vacuum chamber was built and lined with the reflective material. The air was pumped out and the chamber was maintained at as near to a vacuum as possible. The heat transfer between the two surfaces was then studied. There is a slight component remaining due to conduction, but the transfer is mainly by radiation, and by tests of that manner I believe that it would be possible to break down the over-all transfer into the component factors. If such basic data were available it would be possible to synthesize any over-all factor that might be needed for other conditions then those for which the tests were made.

Another point on coating the surfaces with lacquer, is that the reflection by polished aluminum is more or less specular in nature, in that all wave-lengths are reflected equally, or approximately so. Therefore should we coat the surface with a lacquer which has selective reflective properties and which also would absorb a small amount of the incident radiation and thereby affect the surface temperature, a different result would be expected. Whether such would be an improvement or otherwise as compared to uncoated aluminum foil under service conditions is open to question.

It was said that $\frac{5}{4}$ power was purely an empirical factor. I recall having read at one time a derivation involving the combination of the hydrodynamic equation and the heat conduction equation in which the $\frac{5}{4}$ power of the temperature difference for convected heat transfer was predicted analytically, so it is not as empirical as some of us might expect.

PROFESSOR WILKES: I have been arguing for years for the break-down between radiation and convection. Prof. C. M. F. Peterson and myself published a paper [11] on just that point where we were trying to get the engineer to separate radiation out from convection because they do not obey the same laws.

The chemical engineer has done it in that he separates the two, but the mechanical engineer and most other engineers still try to make some simple formula apply to both of these when they do not obey the same laws.

We did not attempt to separate them out in this paper, as over-all coefficients were reported. If you will refer to the paper [12] you will find this matter was discussed thoroughly.

The question involving lacquering of the foil surface brings up a point that should be treated very carefully. Early in the development of aluminum foil insulation I wrote to a manufacturer for some lacquer to put on foil, as I wanted to find out what it would do. I was advised that the company would be very glad to furnish lacquered foil, but they did not think I would be able to lacquer it, and I think they were probably right. It is a very tricky proposition and it certainly ought to be put on by the manufacturer. You cannot take any ordinary transparent lacquer and get the desired result.

In regard to the question of the difference in the reflection from a lacquered surface and a plain surface for various wave-lengths, the emissivity that is determined is total emissivity (all wave-lengths) at low temperature, so it takes account of that without any trouble.

I am glad that $\frac{5}{4}$ power theoretical derivation was mentioned because I have always assumed that it was a purely experimental determination.

[11] Radiation and Convection from Surfaces in Various Positions, by Gordon B. Wilkes and Carl M. F. Peterson. (ASHVE TRANSACTIONS, Vol. 44, 1938, p. 513.)
[12] Loc. Cit. Note 11.

J. D. Edwards: When lacquer is applied to foil in manufacture, it is applied by a roller coating process. The amount of lacquer applied is under very close control. As a result, in commercial operations it has been possible to apply lacquer and maintain the emissivity below a value of 10 per cent regularly. The lacquer applied is of such protective nature that it can stand continuous immersion in water for long periods without any attack on the foil.

Only a few conditions require the use of lacquered foil. For example, in a refrigerator where the insulation is not sealed, and where moisture can condense and leave water in contact with the foil continuously, it is desirable to have such a protective coating on the foil. In most insulated structures, however, the amount of moisture condensing is not sufficient to necessitate any such protection.

The authors' data apply to the conditions met with in building construction and should facilitate the practical application of aluminum foil insulation.

PERFORMANCE OF A STOKER-FIRED WARM-AIR FURNACE AS AFFECTED BY BURNING RATE AND FEED RATE

By A. P. KRATZ* AND S. KONZO,** URBANA, ILL.

INTRODUCTION

THE investigations in forced-air heating in the Research Residence at the University of Illinois during the heating season of 1937-1938 were confined to studies [1] of the performance of a warm-air furnace, fired both by hand and by means of an underfeed stoker.

These tests were made with a high volatile bituminous coal obtained from Saline County (5th vein), Illinois. In the stoker-fired plant, shown in Fig. 1, the initial tests were made with the coal feed rate maintained at 48.6 lb per hour and with the quantity of air for combustion adjusted to maintain a burning rate of approximately 18 lb per hour during the *on-periods* of the stoker. For the heating season of 1938-1939, these investigations were extended to include the performance with the feed rate reduced from 48.6 lb per hour to 27 lb per hour and the burning rate maintained at 18 lb per hour; and with the feed rate maintained at 27 lb per hour and the burning rate reduced from 18 lb per hour to 13 lb per hour. In order to obtain comparable data at all burning rates and feed rates no changes were made in the furnace, in the plant, in the volume of air circulated, nor in the settings of the thermostats. Furthermore, in all of the tests made with the stoker, 1 in. x 10 mesh stoker coal, washed and oil treated, was used.

DESCRIPTION OF THE RESEARCH RESIDENCE AND HEATING EQUIPMENT

The Research Residence in Urbana, Illinois, and the forced-air heating plant have been completely described in a previous publication.[2] The residence is

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[1] Performance of Stoker-Fired and Hand-Fired Warm-Air Furnaces in the Research Residence, by A. P. Kratz, S. Konzo and R. B. Engdahl. (ASHVE TRANSACTIONS, Vol. 45, 1939, p. 297.)
[2] University of Illinois, *Engineering Experiment Station Bulletin No. 266*, by A. P. Kratz and S. Konzo, 1934.

a three story structure of standard frame construction. At the time that these tests were made the walls were not insulated. The double-hung windows were not equipped with weather strips, and during the tests they remained tightly locked.

The heating plant consisted of a cast-iron, circular-radiator warm-air furnace used in connection with a forced-air heating system. The principal dimensions are shown in Table 1. A centrifugal fan delivered 1675 cu ft of air per minute through a duct system consisting of two main warm air trunks and three cold air returns.

As shown in Fig. 1, the stoker was of the underfeed type and was inserted through the ash pit door of the furnace. A balanced check damper was installed in the clean-out of the chimney. The hopper had a capacity of 300 lb,
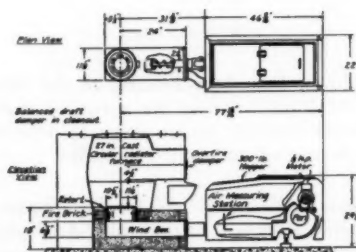
FIG. 1. DIAGRAM OF CONVERSION STOKER INSTALLATION IN FURNACE. RESEARCH RESIDENCE, SEASON 1937-1938, SERIES 3-37 AND 4-37

and coal was delivered to the retort by means of a rotating screw. Both the rate of fuel input and the rate of air supplied to the tuyeres could be independently adjusted. A Pitot-tube, shown at the measuring station in Fig. 1, was used to compute the quantity of air from observations of the velocity at different points in the cross-section of the air tube, and a Wahlen gage was used to measure the velocity pressures.

The heating plant was controlled by a room thermostat operating to start and stop the stoker motor, and to start and stop the circulating fan. The room thermostat was of the heat-anticipating type, and was used in conjunction with two bonnet thermostats [3] which served as high and low limit controls for the temperature of the air in the furnace bonnet.

METHOD OF CONDUCTING TESTS

The average of the air temperatures in all of the rooms of the house was maintained at 72 F both day and night. Observations of weather, indoor room

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[3] A complete wiring diagram of the control system is given in the previous paper. Loc. Cit. Note 1.

air temperatures, room relative humidities, and other incidental data were made daily at 7:00 a.m., 11:00 a.m., 4:00 p.m., and 10:00 p.m. Complete data were obtained for each 24-hour test period on the fuel consumption, weight of ash and clinkers removed, the total integrated time of operation of the fan and of the stoker, the total electrical input of the fan and of the burner motors, and the total number of the *on-periods* of both the circulating fan and stoker. Daily observations were made of the volume of air circulated, and the filters were cleaned with sufficient frequency to maintain the air volume constant. In addition, continuous records of temperatures, CO_2, and the index of smoke density were obtained for each 24-hour period. For each series of tests, data were obtained over a wide range of outdoor weather conditions. For all of the tests made during the two heating seasons with the mechanical stoker,

TABLE 1. DIMENSIONS AND AREAS OF FURNACE USED

Grate diameter	23 in.
Grate area	2.88 sq ft
Firepot diameter	27 in.
Heating surface	
ash-pit	1.36 sq ft
fire pot	8.10 sq ft
dome	18.87 sq ft
radiator	31.93 sq ft
Total	60.26 sq ft
Ratio of heating surface to grate area	20.9
Casing diameter	50 in.
Free area through casing	4.97 sq ft
Combustion space [a]	8.8 cu ft
Free area through over-fire damper	4.9 sq in.

[a] Combustion space is defined in this case as the space above the hearth level, including the dome, but not the feed neck.

each day at 11 a.m., the clinkers were removed, the fuel bed was levelled, and the hopper was filled with coal. During extremely mild weather no attention was given the fuel bed or hopper, except as required every two or three days. By means of the balanced check damper the draft in the smoke pipe was maintained at approximately 0.05 in. of water.

The following is a description of the different test series, and in every case both the feed rate and the burning rate refer to the rates existing during the *on-periods* of the stoker:

Feed Rate 48.6 lb per hour, Burning Rate 18 lb per hour. Overfire Damper Open (*Series 3-37*): In this series of tests, which was discussed in the previous paper,[4] a high feed rate was used in conjunction with a low burning rate, and the overfire damper in the firing door was open at all times. During the *on-periods* of the stoker the CO_2 content of the flue gas was maintained at approximately 10 to 11 per cent, and a slightly hazy atmosphere over the fuel bed was obtained.

Feed Rate 48.6 lb per hour, Burning Rate 18 lb per hour. Overfire Damper Closed (*Series 4-37*): In the first series of tests, *Series 3-37*, considerable burning, accompanied by low CO_2 content of the flue gas occurred during the *off-periods of the stoker*.

[4] Loc. Cit. Note 1.

Hence a parallel series of tests, designated as *Series 4-37* and discussed in the previous paper, was conducted with the overfire damper closed.

Feed Rate 27.0 lb per hour, Burning Rate 18 lb per hour.   Overfire Damper Closed (Series 1-38, 2-38, 3-38):   For the tests conducted during part of the heating season of 1938-1939, the feed rate was adjusted to 27.0 lb per hour and the burning rate was maintained at 18 lb per hour.   Thus the feed rate was lower than the corresponding rate maintained in the tests conducted during the heating season of 1937-1938, while the burning rate was the same.

Feed Rate 25.8 lb per hour, Burning Rate 13 lb per hour.   Overfire Damper Closed (Series 4-38):   A series of tests was conducted during the heating season of 1938-1939 with the feed rate maintained about the same as that used in the preceding series, but with the air input to the stoker reduced to maintain a burning rate of approximately 13 lb per hour.

RESULTS OF TESTS

Determination of Burning Rate

In the case of uniform fuel bed such as that obtained in a hand-fired furnace, the burning, or combustion rate is dependent only on the quantity of air passing through the fuel bed.   Kreisinger [5] has indicated that for uniform fuel beds of this nature approximately one half of the air required for complete combustion must be supplied over the fire.   Hence, if at any given time the weight of air supplied to the grates and the amount of overfire air are known, an observation of the CO_2 content of the flue gas used in connection with the analysis of the fuel affords a means of calculating the instantaneous rate of burning of any specific fuel.   If this rate remains fairly constant, a few observations of the CO_2 will be sufficient to determine a representative mean rate of burning.

The burning rate in the case of a stoker-fired furnace is not necessarily determined by the rate of coal being fed to the furnace.   On the contrary, as indicated in the previous paper,[6] good results were obtained with an intermittently fired, warm-air furnace utilizing a high volatile Illinois coal at an intermittent feed rate of 48.6 lb per hour accompanied by an average burning rate of about 18 lb per hour.   The burning rate always refers only to the *on-periods* of the stoker, since no accurate determinations of air weights or burning rates could be made during the retarded combustion accompanying the *off-periods* of the stoker.

As shown in Fig. 2, in a non-uniform fuel bed, such as that obtained in a furnace fired by means of an underfeed stoker, the air supplied through the tuyeres is utilized for both *primary* and *secondary* combustion.   That is, most of the air supplied through the inner tuyeres, *a,* passes through a fuel bed composed of unburned coal and incandescent coke; whereas practically all of the air supplied through the outer tuyeres, *b,* and part of the air supplied through the inner tuyeres, *a,* passes through a porous fuel bed consisting of incandescent coke, clinkers, and loose ash.   It follows that the combustion rate is determined by the amount of air that comes into contact with the incandescent coke, all the rest acting as secondary air.

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[5] Low Rate Combustion in Fuel Beds of Hand Fired Furnaces, by Henry Kreisinger, C. E. Augustine and S. H. Katz. (*U. S. Bureau of Mines Technical Paper* No. 139, p. 7.)
[6] Loc Cit. Note 1.
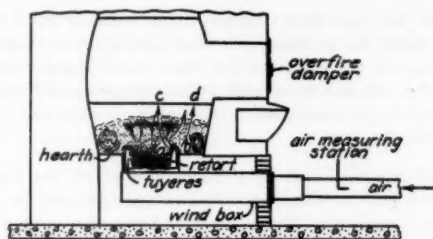
FIG. 2.   FLOW OF AIR THROUGH STOKER FUEL
BED

The diagrams in Fig. 3 showing the distribution of CO_2, O_2, and CO over the fuel bed were obtained by traversing the combustion space over the fuel bed with a portable sampling tube made of fused silica and by analyzing the gases in the orsat gas analyzer.   These diagrams indicate that the flue gases in that portion of the fuel bed directly above the retort contain appreciable percentages of CO, as a result of the primary combustion that occurs as the air passes through the relatively dense layer of incandescent coal and coke predominant in the middle of the fuel bed.   Part of the air which passes through the outer tuyeres and enters the loosely packed layer of incandescent coke, clinkers, and ash surrounding the middle of the fuel bed, escapes to the surface without coming into contact with the incandescent fuel, as indicated in Fig. 3 by the high concentration of O_2 and the low concentrations of CO and CO_2 in that region.   The combustion over the fuel bed is completed when the *secondary* air passing through the outer zone of the fuel bed comes into contact with the unburned gases passing through
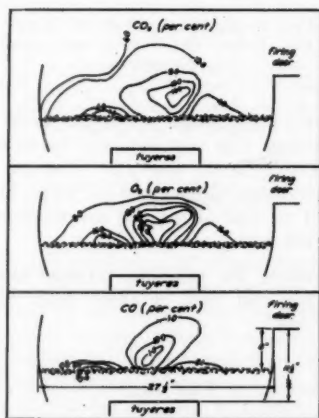
FIG. 3.   DISTRIBUTION OF CO_2, O_2,
and CO

the center zone of the fuel bed. Hence, in a furnace fired by means of an underfeed stoker, both the *primary* air and *secondary* air enter the fuel bed through the tuyeres in the firepot. In most installations, therefore, it may be expected that the admission of additional *secondary* air through the firing door is not necessary for combustion during the *on-periods,* and results only in a dilution of the flue gases.

As with the uniform fuel bed in a hand-fired plant, the burning rate in a stoker-fired fuel bed can be obtained for any specific coal by determining the total quantity of air supplied to the fuel bed and the CO_2 content in the flue gas. If no additional air is admitted through the firing door, the total air supplied to the fuel bed can be determined by measurements made in the air tube
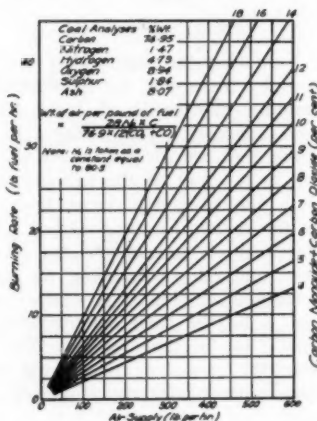
FIG. 4. RELATION BETWEEN BURN-
ING RATE, $CO_2 + CO$, AND AIR
SUPPLY

leading to the wind box of the stoker. For the tests made in the Research Residence plant the burning rate was obtained by using these measurements in conjunction with the CO_2 in the flue gas. For the purpose of measuring the quantity of air, the overfire damper and all joints in the wind box and the firing door, were sealed so that all of the air required for combustion was supplied through the wind box.

Fig. 4 is a chart showing the relations between burning rate, air supply, and CO_2 content for the coal used in these tests. Similar charts can be derived for any other fuel.

Burning Rate and Thickness of Fuel Bed

To a certain extent the frequency and duration of *on-periods* of the stoker depend on the type of heating system in which the stoker is used. In boiler

practice the stoker is usually operated infrequently and the *on-periods* are relatively long, whereas in warm-air furnace installations the converse is true. This fundamental difference in the frequency of *on-period* operation for boilers and for furnaces should be appreciated in order to obtain a clearer understanding of the relations that exist between the burning rate, the feed rate, and the thickness of the fuel bed.

During relatively long *off-periods* of the stoker, the combustion that occurs throughout the period reduces the amount of combustible in the fuel bed,
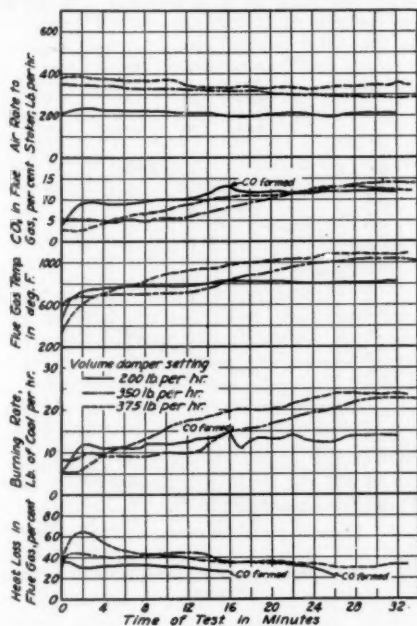
FIG. 5.   COMBUSTION CHARACTERISTICS FOR
CONTINUOUS STOKER OPERATION (TESTS PRE-
CEDED BY LONG OFF-PERIOD)

with the result that both the percentage of combustible matter and the thickness of the fuel bed are decreased. Hence, at the beginning of a stoker operation, following a long *off-period*, the flue gas temperature, the CO_2 content in the flue gas and the burning rate are all at minimum value. After the stoker has been operating for a long period of time, the percentage of combustible in the fuel bed and the thickness of the fuel bed both increase. The result is that the air passes through a thicker layer of incandescent coke, and the CO_2 content of the flue gas is increased.

The graphical log in Fig. 5 shows the results obtained in the Research

Residence plant when the stoker was manually operated so that a long *off-period* was succeeded by a long *on-period*. This condition occurs in the operation of a warm air furnace plant only during the morning pick-up following the operation with a reduced thermostat setting at night. With a uniform fuel bed and a constant amount of air supplied to the grates, the combustion rate remains constant, and a few observations of the CO_2 will be sufficient to determine the average combustion rate. In a stoker-fired plant in which the feed rate is greater than the combustion rate, during long *on-periods* a progressive change occurs in the amount of incandescent fuel with which air comes into contact, resulting in a progressive change in the combustion

No.	Feed Rate	Burning Rate, apx.	Overfire Damper	Series No.
1	27.0	18	Closed	1,2,3,3-38
2	48.6	18	Open	3-37
3	25.8	13	Closed	4-38
4	48.6	18	Closed	4-37
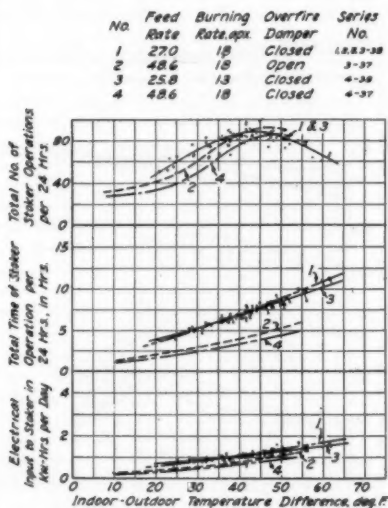
FIG. 6. OPERATING CHARACTERISTICS OF STOKER

rate, and a large number of CO_2 observations are necessary to determine the progress of the combustion rate.

Because in these tests the air rate was maintained constant by means of the constant volume damper in the stoker air tube, the increase in the CO_2 content, as shown in Fig. 4, was reflected in an increase in the burning rate. For extremely long periods of stoker operation the increases in CO_2, flue gas temperature, and burning rate continued until the burning rate finally approximated the feed rate. Because the coal was being burned as fast as it was fed to the furnace, the thickness of the fuel bed tended to remain constant, and the combustion may be considered as having attained uniformity with respect of time. Under some conditions with prolonged *on-period* operation, and with low rates of air input the thickness of the fuel bed built up to such an extent that smoky combustion, accompanied by measurable amounts of

CO, was obtained. This unfavorable combustion was aggravated when the resistance of the fuel bed surrounding the firepot was excessively large, such as that which occurred when large dense clinkers were formed.

From the standpoint of ideal thermostatic control it is desirable to provide continuous combustion just sufficient at all times to offset the heat losses from the house at all outdoor temperatures. Continuous and modulated combustion is usually not obtained in an intermittently fired heating plant, because the burning rate is adjusted to a maximum value sufficient to heat the house under extreme weather conditions, and the regulation of the combustion rate to satisfy milder heating demands is obtained by intermittent operation of the burner. In the case of a forced air furnace plant, owing to its flexibility and lack of heat storage capacity, long and infrequent operations of the stoker are not conducive to the maintenance of uniform bonnet air temperatures. For this reason the *on-periods* of the stoker should be frequent and of short duration, and active combustion occurring during the *off-periods* serves a useful purpose by tending to minimize the differences between the average rate of combustion and the rates taking place during the *on-periods* and the *off-periods*.

From tests made in the Research Residence it was observed that under normal conditions when the stoker was operated frequently and for short periods, the variations in the flue gas temperatures, in the CO_2 content, and in the burning rate, between the maximum occurring during the *on-periods* and the minimum occurring during the *off-periods*, were not as large as those indicated in Fig. 5, which were characteristic of excessively long *on-periods*. Within one or two minutes after the stoker started, the burning rate attained a value which remained constant until the room temperature rose to normal and the stoker was stopped. This burning rate was less than the feed rate, and in the tests reported in the previous paper,[7] the combustion occurring in the *off-periods* consumed the surplus coal fired into the fuel bed, with the result that the thickness of the fuel bed remained constant during the entire 24-hour test period, even though the feed rate and the burning rate differed considerably. Obviously wide discrepancies between feed rate and burning rate are practical only when the coal can burn freely during the *off-periods* of stoker operation. This would not be true for a stoker having an automatic cut-off in the air tube or when using a strongly coking coal. For a coal which tends to form coke trees, it is probable that the air input should be adjusted so that the burning rate is approximately equal to the feed rate.

Variations in Feed Rate and Burning Rate

The results obtained from the tests made to determine the effect of varying the feed rate and burning rate in an intermittently operated furnace plant are shown in Figs. 6 to 9. Significant values for each 24-hour period were plotted against the difference in temperature existing between the indoors and outdoors for the same period. The plotted points deviate to some extent from the curves representing the average of the observed data, and these deviations can be partly attributed to the wind and sun effects which cannot be represented on curves based on temperature difference alone. For purposes

[7] Loc. Cit. Note 1.

of comparison the curves representing the data obtained during the season of 1937-1938 have been transferred to Figs. 6 to 9, and are designated as curves numbers 2 and 4.

As shown in Fig. 6 the total time of stoker operation per day, and the electrical input to the stoker motor per day, were less for high feed rates than for low feed rates.   The unit electrical inputs were approximately 7.8
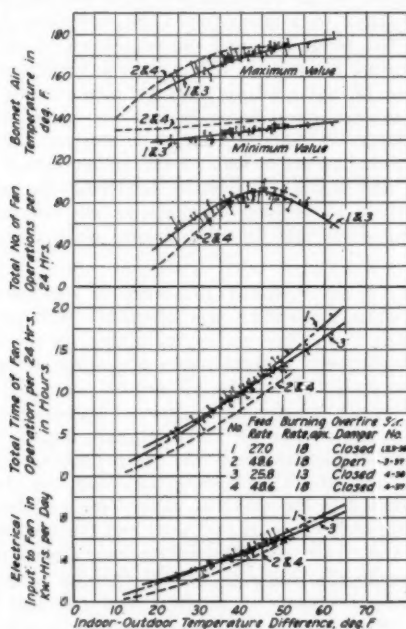
FIG. 7.   OPERATING CHARACTERISTICS OF FAN

and 12.9 kwhr per ton for feed rates of 48.6 lb per hour and 26 lb per hour respectively.

The total time of operation of the circulating fan in the forced air system, and the electrical input to the fan motor, as shown in Fig. 7, were also less for high feed rates than for low feed rates.   With the higher feed rates, due to a large amount of *off-period* burning, higher average bonnet air temperatures were maintained, as shown in the top part of Fig. 7.   As a result of the higher register air temperatures accompanying the higher bonnet air temperatures, shorter periods of fan operation were required in order to cause the room temperature to rise a sufficient amount for the room thermostat to stop the fan and stoker.

The curves in Fig. 8 indicate that minimum average flue gas temperatures

and minimum fuel consumption were obtained with a 48.6 lb per hour feed rate, and 18 lb per hour burning rate, and with the overfire damper closed (Curve No. 4). This arrangement was not entirely satisfactory since a smoky flame was obtained during the *off-periods,* thus indicating that with a thick fuel bed, sufficient secondary air was not being admitted through the outside tuyeres, and that it was necessary to augment the *secondary* air by opening the overfire damper. When the overfire damper was open, the
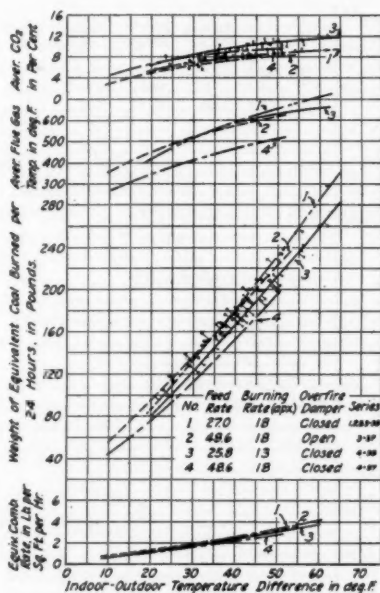
FIG. 8. PERFORMANCE CURVES FOR STOKER-
FIRED FURNACES

smokiness during the *off-periods* was eliminated but the flue gas losses during the *off-period* were larger, resulting in increased fuel consumption, as shown by Curve No. 2. When the same burning rate of 18 lb per hour was maintained and the feed rate was reduced to 27 lb per hour, (Curve No. 1) a thinner fuel bed was obtained. Owing to the evolution of less volatile gases during the *off-period,* the *secondary* air admitted to the fuel bed through the outside tuyeres was sufficient to eliminate smoke without the admission of air through the overfire damper. The thinner fuel bed was also accompanied by less *off-period* burning and more intense combustion over a smaller area during the *on-period,* thus resulting in higher flue gas temperatures which were reflected in greater fuel consumption than that obtained with the

thicker fuel bed under comparable conditions, as shown by Curves Nos. 1 and 4 respectively. There was a greater tendency for the formation of holes in the fuel bed and a greater amount of fly ash was obtained. Therefore, taking all things into consideration, the use of the smaller feed rate cannot be regarded as advantageous for the coal used.

When the burning rate was reduced to 13 lb per hour, which was about enough to take care of the maximum heating demands of the house, and the
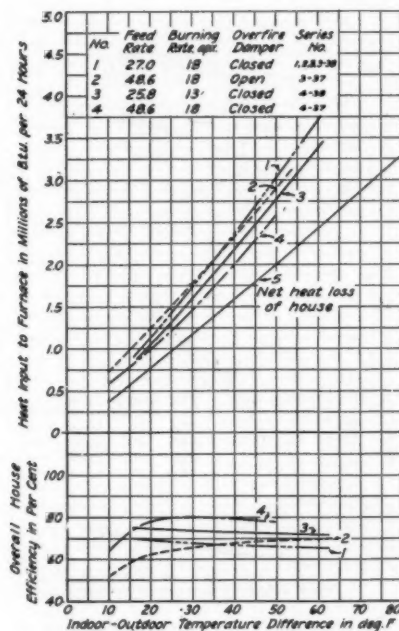
Fig. 9. Heat Inputs to Research Residence

feed rate was maintained at about 26 lb per hour (Curve No. 3), the resulting fuel consumption was less than that obtained with an 18 lb burning rate (Curve No. 1). These results confirm the conclusion that minimum fuel consumption was obtained when the maximum combustion rate most nearly conformed to the maximum heating demands of the house, which was drawn in connection with the studies of oil burning [8] in the Research Residence. In the case of the tests represented by Curve No. 3, the *on-period* burning rate was maintained at a low value and, as far as conditions would permit in this type of plant, the operation of the stoker was nearly continuous. On the other hand, for the tests represented by Curve No. 4, in which

[8] Performance of Oil-Fired, Warm-Air Furnaces in the Research Residence, by A. P. Kratz and S. Konzo. (ASHVE Transactions, 1937, Vol. 43, p. 215.)

considerable burning occurred during the *off-periods*, the average of the combustion rates during the *on-periods* and the *off-periods* was maintained at a minimum value and the *on-periods* were comparatively short.

The fuel quantities shown in Fig. 8 were reduced to terms of heat input to the furnace, in millions of Btu per 24 hours and were plotted as shown in Fig. 9. As explained in the previous paper,[9] the net heat loss from the house (Curve No. 5) was derived from experimental results for heat input obtained with anthracite as fuel. By using these derived values of the net heat loss from the house in connection with the fuel consumption curves for the stoker-fired plant, the over-all house efficiency, or the ratio of the heat loss from the house to the heat input to the furnace, could be calculated and is shown in the lower part of Fig. 9. For an indoor-outdoor temperature difference of 34 F the over-all house efficiencies ranged from 67 to 80 per cent, with an average of about 70 per cent. This range in efficiencies was reasonably small considering the very large differences in the burning rates and feed rates that were used in these tests. It is apparent that an intermittently operated stoker in a forced-air heating plant is extremely flexible, and that a wide range of adjustments in the air rate and feed rate will give acceptable combustion conditions.

Conclusions

The following conclusions may be considered as applying to the Research Residence and the conditions under which the tests were conducted.

1. The burning rate in the case of a stoker-fired furnace is not necessarily determined by the rate of coal being fed to the furnace, but rather by the rate at which the air is supplied to the fuel bed.

2. In the case of a domestic underfeed stoker, the air supplied to the fuel bed through the tuyeres is utilized for both primary and secondary combustion.

3. With a constant air supply to the fuel bed, the burning rate increases progressively during long *on-periods* of the stoker. During short *on-periods* the burning rate remains comparatively constant.

4. Any combustion occurring in the *off-periods* reduces the surplus coal fired into the fuel bed, with the result that the thickness of the fuel bed tends to remain constant, even though the feed rate is considerably in excess of the burning rate.

5. With a freely burning coal that does not coke strongly, a feed rate only slightly in excess of the burning rate results in a thinner fuel bed and a greater fuel consumption than those obtained with a feed rate considerably in excess of the burning rate.

6. The amount of smoke obtained during the *off-periods* may limit the amount that the feed rate can be in excess of the burning rate.

7. An intermittently operated stoker in a forced-air heating plant is extremely flexible, and a wide range of adjustments in the air rate and feed rate will give acceptable combustion conditions.

Acknowledgments

CALCULATION OF COIL SURFACE AREAS FOR AIR COOLING AND DEHUMIDIFICATION

By JOHN MCELGIN* AND D. C. WILEY,** PHILADELPHIA, PA.

I N recent years the problem of simultaneous cooling and dehumidification of air with cold surfaces has received the attention of a considerable number of experimenters. The results of these studies have been to evolve certain concepts and fundamental expressions that relate in an instantaneous manner the variables in this particular heat flow mechanism. Of these, the concept of surface temperature, Lewis'[1] straight line law, Merkel's[2] derivation ot the proper heat flow potential, and Goodman's[3] derivation of the condition curve are outstanding and are widely accepted as basic in the industry.

It is a major purpose of the authors to develop from this background a rational and convenient method of anticipating the area of fin and tube surface required to remove from air a specific total heat quantity in which both sensible and latent heat are transferred. Heretofore, one of the difficulties in the application of fundamental expressions to accomplish this has resulted from the complex equations necessary to relate total heat and temperature at saturation over a wide range of temperature. The method to be presented, by employing a straight line relation between these two quantities over a narrow range, arrives at a final expression in which the required area is determined directly as a function of the mean difference between the total heat of the air and total heat corresponding to saturation at surface temperature. Of further interest is the presentation of a surface temperature chart that permits a graphical solution for the condition at which dehumidification begins for that type of problem in which a portion of the surface is in a dry condition.

Throughout the paper consideration is given only to the counterflow type of problem employing water as the cooling medium. It will be evident later that the method derived is also applicable to cases involving a constant temperature refrigerant and to parallel flow.

In order to employ the essential concept of surface temperature, it is necessary to idealize fin and tube surface as shown in Fig. 1 to consist of a resistance

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** Research Engr., John J. Nesbitt, Inc. ASSOCIATE MEMBER ASHVE.
[1] *Industrial and Engineering Chemistry*, C. S. Keevil and W. K. Lewis, Vol. 20, No. 10, October, 1928, p. 1058.
[2] Merkel, Zeit, Ges. Kalte Industrie, Vol. 34, July, 1927, p. 117.
[3] Dehumidification of Air with Coils by William Goodman (*Refrigerating Engineering*, Vol. 32, No. 4, October, 1936, p. 225).

Presented at the 46th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Cleveland, O., January, 1940.

to heat flow which is constant for a given surface. This permits the assumption that the temperature of the surface is constant at any point in a plane through the coil at right angles to the air flow. For a given water velocity the resistance to heat flow of the water film is also constant and the sum of the fin, tube and water film resistances may be expressed as a constant R.

Between the air and the surface is a film of stagnant air offering resistance to heat flow. This resistance will be expressed in reciprocal form as $1/f_g$ where f_g is the airside coefficient of sensible heat transfer. The value of f_g is assumed constant for a given air velocity.

The mathematical and experimental works previously mentioned have indicated that the droplets of water condensed from the air on the surface have
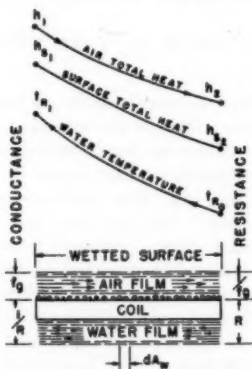
FIG. 1. IDEALIZATION OF FIN
AND TUBE TYPE SURFACE

little, if any, effect on either f_g or R. In this paper it is assumed that surface moisture does not affect f_g or R.

The transfer of heat from the warm moist air to the cold surface and thence through the fins and tubes to the cold water may best be analyzed by dividing the process into two distinct steps, (1) from the air to the wetted surface and (2) from the surface through the fins and tubes to the water.

The total heat lost by the air in passing over an element of wetted area dAw is given by

$$dH_r = \frac{f_g}{s}(h - h_a)\, dAw \dots\dots\dots\dots\dots\dots\dots\dots\dots (1)$$

This equation which was originally derived by Merkel,[4] combines sensible heat transfer due to temperature difference and latent heat transfer due to vapor pressure difference into a single equation which states that the rate of simultaneous sensible and latent heat flow depends upon the difference between the total heat of the air flowing over the surface and the total heat corresponding to saturation at the surface temperature.

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[4] Loc. Cit. Note 2.
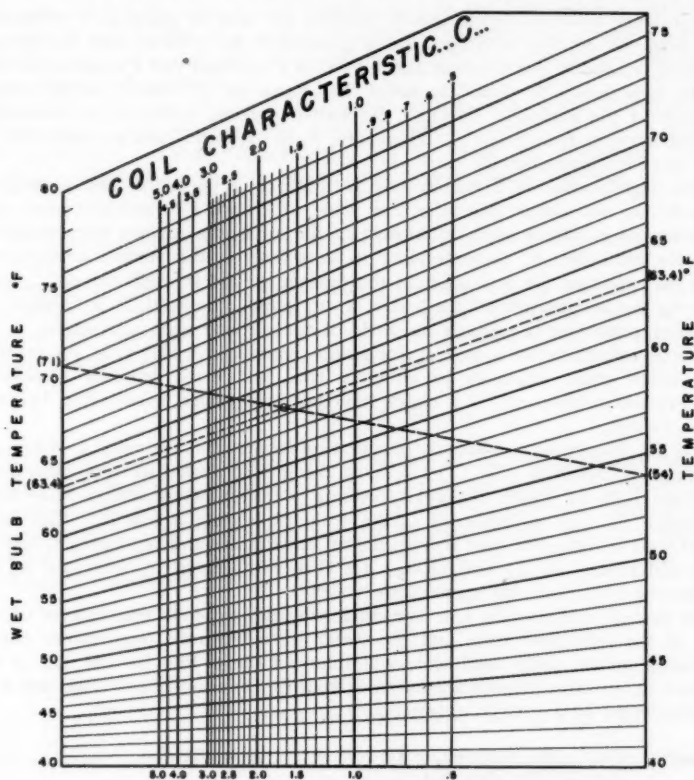
FIG. 2. SURFACE TEMPERATURE CHART

The total heat transferred from the element of area dAw through the fins and tubes to the water is given by

$$dH_T = \frac{1}{R}(t_a - t_n)\, dAw \dots\dots\dots\dots\dots\dots\dots\dots\dots(2)$$

Where R is as defined above the composite resistance to heat flow imposed by the fins and tubes and the internal water film.

Equating (1) and (2) and rearranging

$$\frac{t_a - t_n}{h - h_n} = \frac{f_g R}{s} = C \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots(3)$$

This is essentially the same form as originally derived by Goodman.[5]

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[5] Loc. Cit. Note 3.

The value of s, the humid specific heat of air, may be taken as constant for the range of comfort cooling. For a given coil, air velocity and refrigerant velocity, f_g and R are constant and thus C is a constant coil characteristic for given conditions. It should be noted that Equation (3) refers to the wetted portion of the surface. Through the wetted portion, moisture is condensing on the surface from the air. Therefore, h_s is the total heat at saturation at the surface temperature t_s.

The significance of Equation (3) is that for a particular coil employing specific air and water quantities, the value of C will be fixed and hence the interdependent values of the total heat of the air h, the surface temperature t_s and the total heat h_s corresponding to saturation at the surface temperature may be evaluated for any point in the passage of air through the coil. Note that t_R, the temperature of the water at any point, is essentially a function of h, being related to it through the ratio of the air and water quantities.

Direct calculation of t_s and h_s for specific values of h and t_R is not feasible and resort must be made to either a trial and error solution employing psychrometric tables or to a chart depicting the proper relationship between these values.

Fig. 2 illustrates a surface temperature chart that permits a graphical solution of Equation (3). The scale at the left is plotted to even increments of total heat to which the corresponding wet-bulb temperatures have been assigned. It may be regarded as the scale expressing h and h_s. The right hand scale is plotted to even increments of temperature and represents t_s and t_R. The differences $(h-h_s)$ and (t_s-t_R) are then represented by intervals on the respective total heat and temperature scales.

As proved in Appendix 1, a line connecting the wet-bulb temperature of the air on the left hand scale and the water or refrigerant temperature corresponding on the right hand scale will intersect the C line for the coil at the prevailing surface temperature as read from the slanting lines. This can best be illustrated by a specific example.

Example 1: For a given surface $f_g = 10$ $\qquad$ $R = 0.04$
$\qquad$ Find (a) the value of C
$\qquad\qquad$ (b) the surface temperature t_s when the refrigerant temperature $t_n = 54$ F and the wet-bulb temperature $t' = 71$ F

Solution: $\qquad$ (a) Using Equation (3) Let $s = 0.245$

$$C = \frac{10 \times 0.04}{0.245} = 1.63$$

$\qquad$ (b) Using the surface temperature chart in Fig. 2 spot the refrigerant temperature $t_n = 54$ F on the right hand scale and the air wet-bulb temperature $t' = 71$ F on the left hand scale. Draw the line intersecting these two points. Where this line intersects the vertical line $C = 1.63$, (interpolate between 1.6 and 1.7) follow the slanting directrix lines to either the right hand or left hand scale and read the surface temperature by interpolation as 63.4 F.

In *Example 1*, the assumption is made that the dew-point temperature of the air at the condition of 71 F wet-bulb is equal to or above the surface temperature 63.4 F; otherwise moisture will not be condensed on the surface and neither Equation (3) nor Fig. 2 will truly represent the actual surface temperature. That a coil may readily operate to remove only sensible heat

from the air for a portion at the entering air side and both sensible and latent heat for the remainder is a matter of common experience. Dehumidification will take place in a coil only where the dew-point temperature is equal to or above the surface temperature.

For the present, discussion will be confined to means of calculating wetted surface areas only, whether this area represents a complete coil or only a portion thereof. Later, consideration will be given the treatment of the problem involving dry surface in combination with wetted surface.

DETERMINATION OF THE WETTED SURFACE AREA REQUIRED

Equation (1), as repeated, states that the potential for the combined sensible and latent heat flow from the air to the wetted surface is the difference between the total heat of the air flowing over the surface and the total heat corresponding to saturation at the surface temperature.

$$dH_\tau = \frac{f_s}{s} (h - h_s) \, dAw$$

Since both h and h_s are variable this expression in the form shown is suitable only for the calculation of the differential area required to remove a differential quantity of heat with the instant potential $(h-h_s)$. To integrate Equation (1) and therefore to sum up the area required to remove from the air a specific total heat quantity, it is first necessary to relate h_s to h.

Restating Equation (3)

$$\frac{t_s - t_n}{h - h_s} = C$$

Rearranging Equation (3) and differentiating

$$Cdh + dt_n = Cdh_s + dt_s \dots\dots\dots\dots\dots\dots\dots\dots (4)$$

The total heat given up by the air in flowing over the surface is at all points equal to the heat gained by the water for the interval considered.

$$W_s dh = - W_n dt_n \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots (5)$$

Substituting Equation (5) in Equation (4) and rearranging

$$\left(C - \frac{W_s}{W_n} \right) dh = Cdh_s + dt_s \dots\dots\dots\dots\dots\dots\dots\dots (6)$$

Equation (6) shows that any relation between h and h_s has its origin in a relation between h_s and t_s, that is, a relation is first required that expresses total heat in terms of temperature at saturation for the range over which the surface temperature varies.

It may be observed either from psychrometric tables or from the total heat temperature line itself that the exact relation between h_s and t_s over a wide range is quite complex. The substitution of such a relationship in Equation (6) so complicates Equation 1 that complete integration is extremely difficult if not impossible.

To arrive at an integrable form of Equation (1) it is, therefore, necessary

to employ a relationship between h_s and t_s that may be accepted as true over a limited range. In Fig. 3, total heat is plotted against temperature at saturation for the range of surface temperatures normally encountered. For limited ranges on this line such as a to b or b to c, a linear relationship between h_s and t_s closely approximates the true values. The degree of accuracy with which the total heat values as given by the chord between the end points checks the actual values is affected by the range employed. For the present, no attempt will be made to define the extent of the acceptable range but attention will be directed toward solution of Equation (1) employing this linear relation.

$$h_s = m + nt_s \qquad \text{for a limited range of } t_s \dots\dots\dots\dots\dots\dots(7)$$

$$\text{Differentiating} \qquad dh_s = ndt_s \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots(8)$$

Substituting (8) in (6)

$$dh_s = \frac{C - \dfrac{W_g}{W_a}}{\dfrac{1}{n} + C}\, dh = Edh \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots(9)$$

This provides the relationship between h and h_s necessary for integration of Equation (1).

As shown in Appendix II Equation (1) may now be integrated to arrive at the final expression for the wetted surface area.

$$A_w = \frac{H_T s}{f_g MHD} \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots(10)$$

Where A_w = wetted surface area required
$\qquad H_T$ = total heat in Btu per hour removed from the air in its passage through the coil
$\qquad\quad = W_g\,(h_1 - h_2)$
$\qquad\qquad h_1$ is initial total heat of the air
$\qquad\qquad h_2$ is final total heat of the air
$\qquad MHD$ = Log mean heat difference between the air and surface.

$$= \frac{\theta_L - \theta_s}{2.3 \; log_{10} \dfrac{\theta_L}{\theta_s}}$$

$\qquad\qquad \theta_L$ = Larger total heat difference between the air and surface
$\qquad\qquad \theta_s$ = Smaller total heat difference between the air and surface.

Example 2: Find the area required to cool air from an entering wet-bulb temperature of 70 F to a final wet-bulb temperature of 55 F using chilled water entering at 45 F and leaving at 55 F.

The air velocity is 500 fpm standard air and the airside coefficient f_g is 9.7. The water velocity is 2 fps and the resistance of the water film and copper R is 0.0378.

Using Equation (3) $\qquad C = \dfrac{9.7 \times 0.0378}{0.245} = 1.5$

Referring to the surface temperature chart as shown in Fig. 2
$\qquad\qquad t_{s1} = 63.0$ F corresponding to $t'_1 = 70$ F and $t_{a3} = 55$ F
$\qquad\qquad t_{s2} = 49.6$ F corresponding to $t'_2 = 55$ F and $t_{a1} = 45$F

$$h_1 = 33.51 \qquad h_{s1} = 28.22 \qquad \theta_1 = 5.29$$
$$h_2 = 23.04 \qquad h_{s2} = 19.97 \qquad \theta_2 = 3.07$$

$$MHD = \frac{5.29 - 3.07}{2.3 \; Log_{10} \dfrac{5.29}{3.07}} = 4.08$$

From Equation (10)

$$A_w = \frac{H_\tau s}{f_s \, MHD}$$

$$H_\tau = W_g \, (h_1 - h_2)$$

$$= \frac{500 \times 60}{13.35} (33.51 - 23.04)$$

$$= 23{,}600 \text{ Btu per hour}$$

$$Then \; A_w = \frac{23{,}600 \times 0.245}{9.7 \times 4.08} = 145.5 \text{ sq ft wetted surface per sq ft of face area.}$$

In the application of the foregoing to specific wetted surface problems there naturally arises the question of the magnitude of the error involved in the calculation of surface areas based on the assumed straight line relationship between h_s and t_s for a particular surface temperature range. This can best be answered by first referring to Fig. 4, that represents the total heat temperature line and the assumed chord to an exaggerated scale.

With air entering a coil at h_1 and leaving at h_2, the corresponding wetted surface total heats will be h_{s1} and h_{s2} respectively and the chord connecting h_{s1} and h_{s2} will represent the straight line assumption between these two end points. It is evident that the total heat as anticipated by the line will (except at the end points) exceed the true total heat for a particular surface temperature by a variable quantity e. The maximum value of e depends on the range over which the assumption is made or approximately on the difference $(h_{s1}\text{-}h_{s2})$. Since the chord lies above the true h_s line, the true $(h\text{-}h_s)$ in Equation (1) is reduced by e. The comparative error introduced in the area required is not only a function of e but depends also upon the potential $(h\text{-}h_s)$. Thus, if $(h\text{-}h_s)$ is small then the deviation between the true and the assumed value of h_s will bear an important relation to the area while if $(h\text{-}h_s)$ is large the significance of the deviation is reduced.

It may be noted that if an intermediate true surface temperature h_{sx} corresponding to an air condition h_x between h_1 and h_2 is established, then calculation of the individual areas for the steps h_1 to h_x and h_x to h_2 will greatly improve the accuracy. The final criterion for any problem is to divide the surface temperature (and air condition) range into a number of individual steps such that the total area representing the sum of the individual areas differs by a negligible quantity from the total area obtained as a result of one less step.

The number of individual steps necessary to provide a degree of accuracy acceptable in engineering calculations is, as implied previously, dependent both upon the range of surface temperatures subtended by the initial and final conditions and the mean heat difference for these conditions.

For surface temperatures between 40 F and 75 F a single solution of Equation (10) will usually check within 3 per cent of a point by point solution
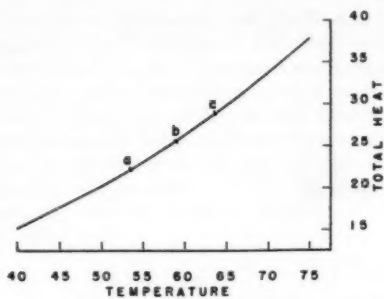
FIG. 3.   TOTAL HEAT VS. TEMPERATURE
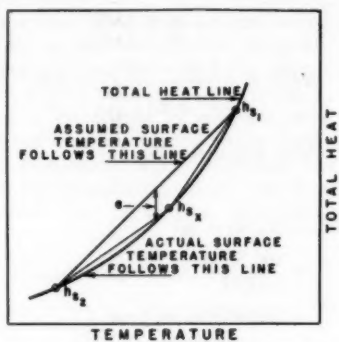
FIG. 4.   EXAGGERATED SCALE OF TOTAL HEAT VS. TEMPERATURE
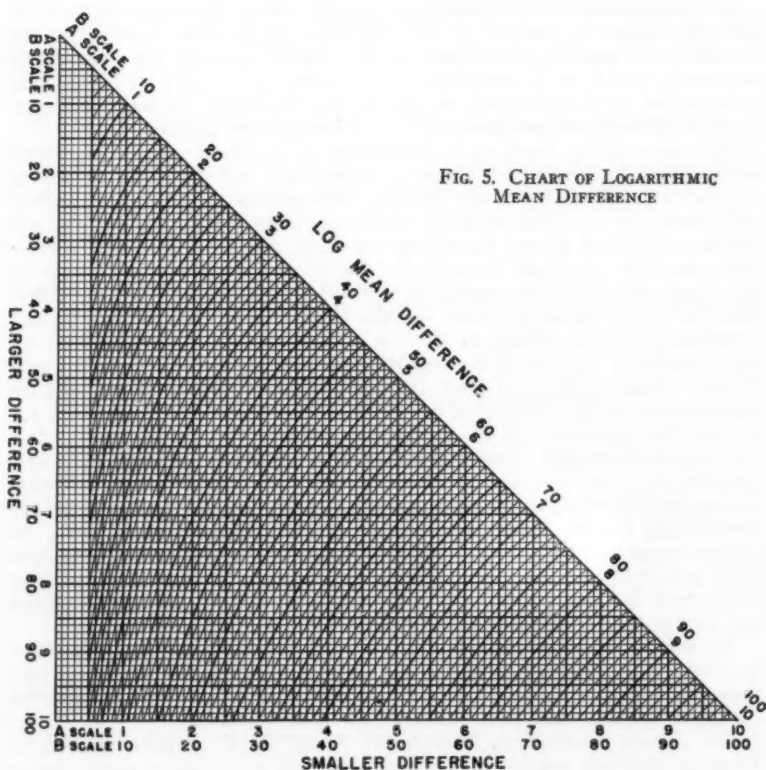
FIG. 5.   CHART OF LOGARITHMIC MEAN DIFFERENCE

where the surface temperature range does not exceed 12 F and the mean heat difference corresponding to the initial and final conditions is greater than 3. Since the chord lies above the true saturation line the error obtained is always of a positive nature. A two step solution will provide approximately the same degree of accuracy for the same mean heat difference over twice the surface temperature range. These boundary conditions are necessarily approximate for it will be apparent that anticipation of the error involved in any problem essentially rests with a multi-point solution.

However, in the range of cooling and dehumidifying problems normally encountered in comfort air conditioning practice a single step solution for the narrower ranges and a two step solution for the wider ranges will prove sufficiently accurate. It only becomes necessary to employ more than a two step solution where the quantity of surface involved is very large and the mean heat difference is of the order of $1\frac{1}{2}$ or less.

Example 2 shows the procedure employed in a single step solution.

The simplest approach to a two step solution is to divide the total heat to be removed in the wetted surface portion of a coil into two equal parts and then to calculate the total area from the respective mean heat differences so established. Thus, if the total heat of the entering air is h_1 and the total heat of the leaving air is h_2, and intermediate total heat h_m is selected such that

$$(h_1 - h_m) = (h_m - h_2)$$

The corresponding surface total heats as determined from the surface temperature chart (Fig. 2) will be h_{s1}, h_{sm} and h_{s2}.

Then combining the expressions for the individual areas as given by Equation (10)

$$A_w = A_{w1} + A_{w2} = \frac{H_T s}{2 f_s} \left(\frac{1}{MHD_1} + \frac{1}{MHD_2} \right) \dots\dots\dots (11)$$

Note that:

$$\frac{H_T}{2} = W_s (h_1 - h_m) = W_s (h_m - h_2)$$

MHD_1 is determined by the conditions h_1, h_m, h_{s1} and h_{sm} while MHD_2 is determined by h_m, h_2, h_{sm} and h_{s2}.

Fig. 5 expressing logarithmic mean heat difference in terms of the larger and smaller differences simplifies the determination of this value.

Example 3: Find the area required to cool air from an entering wet-bulb temperature of 78 F to a final wet-bulb temperature of 55 F using chilled water entering at 45 F and leaving at 60 F.

The air velocity is 300 fpm of standard air and the corresponding air side coefficient $f = 6.48$.

The water velocity is 2 fps and the resistance of the water film and copper, R, is 0.0378.

Using Equation (3) $\qquad C = \frac{6.48 \times 0.0378}{0.245} = 1$

For a two step solution a point midway between h_1 and h_2 is assumed. Since $h_1 = 40.64$, $h_2 = 23.04$, $h_m = 31.84$ and t'_m corresponding is 67.9.

The corresponding water temperature $t_{sm} = \frac{45 + 60}{2} = 52.5$ F

From the surface temperature chart,

$t_{s1} = 68.4$ F corresponding to $t'_1 = 78$ F and $t_{s3} = 60$ F
$t_{sm} = 58.9$ F corresponding to $t'_m = 67.9$ F and $t_{sm} = 52.5$ F
$t_{s3} = 48.6$ F corresponding to $t'_3 = 55$ F and $t_{s1} = 45$ F
$h_1 = 40.64$ $h_{s1} = 32.24$ $\theta_1 = 8.40$
$h_m = 31.84$ $h_{sm} = 25.46$ $\theta_m = 6.38$
$h_2 = 23.04$ $h_{s3} = 19.44$ $\theta_2 = 3.60$

Referring to Fig. 5. MHD_1 between 8.40 and 6.38 is equal to 7.35 and the MHD_2 between 6.38 and 3.60 is equal to 4.86.
H_T per sq ft of face area is equal to

$$300 \times \frac{60}{13.35} \times (40.64 - 23.04) = 23,800 \text{ Btu per hour}$$

From Equation (11)

$$A_w = \frac{23,800 \times 0.245}{2 \times 6.48} \left(\frac{1}{7.35} + \frac{1}{4.86} \right) = \quad \begin{array}{l} 154 \text{ sq ft of wetted surface} \\ \text{per square foot face area.} \end{array}$$

Note: If the above example were solved using but one step, the MHD between the end points $\theta_1 = 8.40$ and $\theta_3 = 3.50$ would be 5.66. Substituting this value in Equation (1)

$$A_w = \frac{23,800 \times 0.245}{6.48 \times 5.66} = 159 \text{ sq ft per square foot of face area.}$$

Or an error of $+ 3$ per cent would have been made if one step were used instead of two steps.

GRAPHICAL DETERMINATION OF THE AIR AND REFRIGERANT CONDITIONS AT THE POINT WHERE DEHUMIDIFICATION BEGINS

If the surface temperature at the point where the air enters a coil is higher than the entering air dew-point temperature, then for a portion of the coil only sensible heat is transferred. The equations for wetted surface derived in this paper apply only to simultaneous sensible and latent heat transfer.
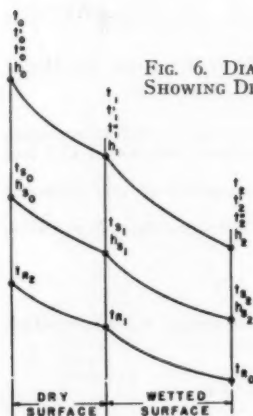
FIG. 6. DIAGRAM OF COIL SURFACE SHOWING DRY AND WETTED SURFACES
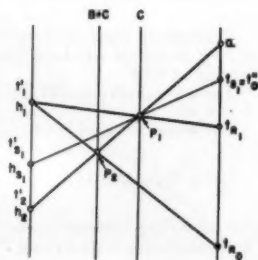
FIG. 7. SKELETON SURFACE TEMPERATURE CHART

FIG. 9. DIAGRAM OF HEAT FLOW FOR WETTED SURFACE

FIG. 8. BASIC SURFACE TEMPERATURE CHART

The well known equations for sensible heat transfer do not apply for combined cooling and dehumidification. As might be expected, large errors must result if the presence of considerable dry surface is neglected and the coil area computed as if all wetted or all dry. The relation between the entering air dew-point temperature and the entering surface temperature should, therefore, be observed early in the solution of a cooling coil problem. If dry surface exists, the coil must be treated as if there were two coils in series; one consisting of all dry surface and the other of all wetted surface. Such a condition is illustrated in Fig. 6.

In order to calculate separately the dry and wetted surface, it is necessary to determine the air and refrigerant conditions at the point where dehumidification begins. The surface temperature at this point is known, that is, it must be just equal to the entering air dew-point temperature in order for dehumidification to just begin. The corresponding air and refrigerant condition may be determined graphically from the surface temperature chart and the psychrometric chart.

The heat lost by the air in flowing over any portion of a coil is equal to the heat gained by the water. Thus referring to the wetted portion of the coil in Fig. 6,

$$W_g (h_1 - h_3) = W_n (tr_1 - tr_0) \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots (12)$$

$$\text{and } \frac{tr_1 - tr_0}{h_1 - h_2} = \frac{W_g}{W_n} = B \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots (13)$$

For any given air and water quantities, B is a constant.
Equation (3) may be partially restated as follows:

$$\frac{f_g R}{s} = C \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots (14)$$

For a given coil, air velocity and water velocity, C is a constant.

Fig. 7 represents a skeleton surface temperature chart. The only slanting

directrix line shown is that of the surface temperature which is equal to the entering air dew-point temperature. The point P_1 on this directrix line is the intersection of the C line from Equation (14). From the point P_1 a line is drawn to t'_2 the leaving air wet-bulb temperature. The value of B from Equation 13 is added to C and the point P_2 determined where the vertical line $B + C$ intersects the line (P_1, t'_2). The entering refrigerant temperature t_{Ro} is spotted on the right hand scale and a line drawn through P_2 to the left hand scale. This point is t'_1, the air wet-bulb temperature at the point where dehumidification begins. From t'_1, a line is drawn through P_1 to the right hand scale. This point is t_{R1}, the refrigerant temperature at the point where dehumidification begins. When the air dew-point and wet-bulb temperatures are known, the corresponding dry-bulb temperature is read from psychrometric chart.

The proof of this graphical method is as follows:

From Fig. 7, $\dfrac{a - t_{a0}}{h_1 - h_2} = B + C$ . . . . (15)

also $\dfrac{a - t_{a1}}{h_1 - h_2} = C$ . . . . (16)

Equating (15) and (16)

$$\frac{t_{a1} - t_{a0}}{h_1 - h_2} = B$$

This is Equation (13).

DETERMINATION OF DRY SURFACE AREA

The amount of dry surface required is determined by the familiar equation

$$A_D = \frac{H_D}{U\,MTD} \quad . . . . (17)$$

Where H_D is the sensible heat lost by the air in flowing over the dry surface and may be calculated from the equation

$$H_D = W_a s\,(t_0 - t_1) \quad . . . . (18)$$

and U is the over-all heat transfer coefficient in Btu per square foot air side area per hour per degree logarithmic mean temperature difference between the air and refrigerant. The value of U is found from the equation

$$U = \frac{1}{R + \dfrac{1}{f_a}} \quad . . . . (19)$$

The mean temperature difference is found by the equation

$$MTD = \frac{D_L - D_s}{2.3\,Log_{10}\dfrac{D_L}{D_s}} \quad . . . . (20)$$

Where $D_L = (t_0 - t_{a2})$ or $(t_1 - t_{a1})$ whichever is larger and
$D_s = (t_1 - t_{a1})$ or $(t_0 - t_{a2})$ whichever is smaller.

CONCLUSIONS

A surface temperature chart is presented which permits the direct determination of surface temperature from the air wet-bulb temperature, the refrigerant temperature and the coil characteristic. A method is presented for the graphical determination on the surface temperature chart of the air and refrigerant conditions at the point where dehumidification begins.

The logarithmic mean total heat difference is derived as the potential causing heat flow from warm moist air to cold wetted surface. The relation between total heat at saturation and temperature is investigated. Equations are presented for computing the amount of surface required to cool and dehumidify air to specified conditions of total heat.

APPENDIX I

THE SURFACE TEMPERATURE CHART

Equation (3) is

$$\frac{t_s - t_R}{h - h_s} = C$$

In Fig. 8, two parallel lines A and B are divided into increments respectively a to the inch and b to the inch. The A scale is marked as total heat and the B scale is marked as dry-bulb temperature. From the B scale, the slanting directrix lines are drawn to the corresponding total heats at saturation on the A scale.

If values of h, h_s, t_s and t_R are spotted on the chart as shown and connected by the lines (h, t_R) and (t_s, h_s), two similar triangles will be formed intersecting at the line C, and from the geometry of the chart

$$\frac{a\,(t_s - t_R)}{b\,(h - h_s)} = \frac{X}{Y} = C$$

Thus lines representing C may be drawn on the chart. In the final form of the chart the total heat values are omitted and only the wet-bulb temperatures are shown on the left hand scale.

APPENDIX II

Derivation of $A_w = \dfrac{H_T \times S}{f_s \times MHD}$

From Equation (1) of the text:

$$dH_T = W_s dh = \frac{f_s}{s}\,(h - h_s)\,dA_w \quad \dots \dots \dots \dots \dots \quad (1)$$

From Equation (9) of the text:

$$dh_s = E\,dh \quad \dots \dots \dots \dots \dots \dots \dots \dots \dots \quad (2)$$

Referring to Fig. 9:

$$\text{Let } \theta = h - h_s \quad \dots \dots \dots \dots \dots \dots \dots \dots \quad (3)$$

then $d\theta = dh - dh_a$ . . . . (4)

Substituting Equation (2) in Equation (4) and rearranging: $dh = \dfrac{d\theta}{1-E}$ . . . (5)

then from Equations (1) and (5)

$$\frac{d\theta}{\theta} = \frac{f_g\,(1-E)}{s\;W_g}\,dA_w\,.$$ . . . . (6)

Integrating between limits of θ_1 and θ_2

$$Log_e\frac{\theta_1}{\theta_2} = \frac{f_g\,(1-E)}{s}\,A_w$$ . . . . (7)

Multiply both sides of Equation (7) by $\dfrac{\theta_1 - \theta_2}{Log_e\dfrac{\theta_1}{\theta_2}}$

$$\theta_1 - \theta_2 = \frac{f_g\,(1-E)}{s\;W_g}\,A_w\,\frac{\theta_1 - \theta_2}{Log_e\dfrac{\theta_1}{\theta_2}}$$ . . . . (8)

Referring to Fig (9)

$\theta_1 - \theta_2 = \triangle h - \triangle h_a$ . . . . (9)
and from Equation (2) $\triangle h_a = E\triangle h$ . . . . (10)

Then from Equations (9) and (10)

$$\triangle h = \frac{\theta_1 - \theta_2}{1 - E}$$ . . . . (11)

Since $H_T = W_g\,\triangle h$ . . . . (12)

$$H_T = W_g\frac{\theta_1 - \theta_2}{1 - E}$$ . . . . (13)

Substituting Equation (8) in Equation (13)

$$H_T = \frac{f_g}{s}\,A_w\,\frac{\theta_1 - \theta_2}{Log_e\dfrac{\theta_1}{\theta_2}}$$ . . . . (14)

By definition $MHD = \dfrac{\theta_1 - \theta_2}{Log_e\dfrac{\theta_1}{\theta_2}}$ . . . . (15)

Then

$$A_w = \frac{H_T \times S}{f_g \times MHD}$$ . . . . (16)

It should be noted that three cases for this derivation exist—

Case 1 where $\theta_1 > \theta_2$ as shown
Case 2 where $\theta_1 = \theta_2$ $MHD = \theta_1 = \theta_2$
Case 3 where $\theta_1 < \theta_2$ The respective positions of θ_1 and θ_2 are reversed.

To cover Cases 1 and 3

$$MHD = \frac{\theta_L - \theta_s}{Log_e \frac{\theta_L}{\theta_s}}$$

where

θ_L = Larger heat difference.
θ_s = Smaller heat difference.

SYMBOLS

A = Total square feet airside area, i.e., the total external surface of the fins and tubes exposed to air flow.

A_D = Square feet dry airside surface.

A_w = Square feet wetted airside surface on which moisture is condensing.

f_g = Sensible heat transfer coefficient through air film in Btu per hour per degree Fahrenheit per square foot airside surface.

W_g = Total weight of air, pounds per hour.

W_n = Total weight of refrigerant, pounds per hour.

H_D = Sensible heat lost by total weight of air flowing over dry surface, Btu per hour.

H_T = Total heat lost by total weight of air flowing over wetted surface, Btu per hour.

h = Total heat content of air vapor mixture per pound of dry air. Btu values are from Goodenough's tables and include the sensible heat of the air plus the heat of vaporization and the heat of superheat of the water vapor. Wet-bulb temperature is taken as the sole index of total heat.

s = Humid specific heat of air vapor mixture, Btu per pound of dry air per degree Fahrenheit.

t = Dry-bulb temperature, degrees Fahrenheit.

t' = Wet-bulb temperature, degrees Fahrenheit.

t'' = Dew-point temperature, degrees Fahrenheit.

R = Resistance to heat flow through fins, tubes and refrigerant film per square foot airside.

U = Over-all sensible heat transfer coefficient. Btu per hour per square foot airside per degree between air and refrigerant.

SUBSCRIPTS

$_0$ = Refers to the initial condition of air or refrigerant and the condition of the surface where the air enters.

$_1$ = Refers to the condition of air, surface and refrigerant at the point where dehumidification begins.

$_2$ = Refers to the final condition of air or refrigerant and the condition of the surface where the air leaves.

$_n$ = Refers to the refrigerant.

$_s$ = Refers to the surface.

$_g$ = Refers to the air.

DISCUSSION

G. L. TUVE: The authors of this paper are to be commended for their simplification of one of the most complex heat transfer problems, and for their clear statements of fundamentals and assumptions. They have accomplished a most difficult task in producing a paper which is brief and yet easy to read. The Research Technical Advisory Committee on Heat Transfer of Surface Coils, at one of its meetings, decided to ask the Research Committee or the Council, to place this entire matter in the hands of the Committee on Standards, so that a Society test code may be produced, dealing with this subject of dehumidifying coils in particular.

This paper illustrates one of the two methods of attack on this complicated problem. The authors have assumed ideal processes and arrangements, such as pure counter-flow, or a straight line relation between total heat and saturation temperature, or a constant surface temperature across any plane at right angles to the flow. This is simplification by assumption and the mathematical analysis is then not too complicated.

The other method of attack, which the authors did not discuss, is that of accounting for deviations between assumed and actual conditions by obtaining factors from tests and experiments. The experimental engineer does not have full confidence in a mathematical analysis until he sees it verified by tests over a wide range.

The choice between a one-step, a two-step or a three-step solution seems to me to be a very good way of using the simple straight-line relations to follow a complex curve. The method as a whole lays much stress on the socalled surface temperature, and since that temperature is one that we can hardly define, we cannot measure and frequently we cannot even picture on a chart. I for one would rather see it passed over lightly or eliminated from the equations, as Mr. Goodman has done.

I should like to ask the authors if they have been able to check experimentally the assumption that the air-side coefficient is not affected by the presence of water drops from dehumidification and if so what about the change in air-contact area when a large amount of condensate is present?

H. B. NOTTAGE:[a] In Equation (1), which is due to Merkel originally and is based upon the assumption that the Lewis relationship between the coefficients of heat and mass transfer is valid, the factor $\frac{F_g}{S}$ is not accurately equal to the transfer factor based upon the enthalpy potential, as was shown by Merkel in his original paper. I do not recall Merkel's exact figures, but I know that an appreciable discrepancy exists.

The author calls this factor h the total heat. I believe that the accepted standard named is enthalpy. The total heat was what is now called the *Sigma Function* by Carrier and others, and is useful only for adiabatic saturation processes.

In the matter of analysis in terms of enthalpy and logarithmic mean enthalpy potential, you might be interested in reading a paper[b] which was presented at the San Francisco meeting of the *American Society of Mechanical Engineers*. In that paper there was developed a similar analysis to the one presented here.

One of the things which might be of further interest is in the interpretation of the fraction of the total surface area which is wetted. We at California found that we could set up an expression for the effectiveness of a cooling tower including

[a] Instructor, Mech. Engrg., Armour Institute of Technology, Chicago, Ill.
[b] Performance Characteristics of a Mechanically-Induced Draft Counterflow Packed Cooling Tower, by A. L. London, W. E. Mason and L. M. K. Boelter. (*ASME Transactions*, January, 1940, p. 41.)

a factor to represent the fraction of the total surface area which is wetted, and we were able to represent that factor in terms of the loading on the tower.

Again, one more comment about this enthalpy potential. It was stated that the enthalpy difference is the truly rational potential for simultaneous sensible and latent heat transfer. I question that. It is a very useful factor to employ for over-all heat and mass transfer balances, but if the system in which heat transfer, evaporation and diffusion all occur simultaneously be treated as a physico-chemical system in which degradation of energy comes about spontaneously in accord with the second law of thermodynamics, it can be shown that the enthalpy is the correct potential for energy flow only when the process occurs at constant pressure and at constant entropy. The condition of constant entropy implies reversibility, or that the temperature differences and the partial pressure differences are infinitesimally small.

Actually, we have finite temperature and partial pressure differences and the approximation involved in using enthalpy as the potential for energy flow becomes greater as the temperature and partial pressure differences increase. It is quite a satisfactory approximation for small differences and enthalpy is very useful as a design variable, but it is not too rational as a potential under all conditions.

S. H. Downs: I would like to know if the authors know the effect of a heavier water film on the condensation, the film being so heavy that it is actually running across the surface of the tubes.

John McElgin: The purpose of this paper has been to develop from two fundamental relations a convenient method of determining wetted surface areas required for cooling and dehumidification. These relations are due to Lewis and Merkel. Merkel's expression for the heat flow potential is given in the text. Lewis derived the relation between the coefficient of vapor diffusion and the coefficient of sensible heat transfer and the use of this results in what is commonly termed the straight-line law, applicable to a constant surface temperature.

It is true that in the derivation of both Merkel's and Lewis' relations certain approximations were made but these are well within the limits required in anticipating the performance of commercial cooling surface. These relations do explain the cooling and dehumidifying process in a much more rational manner than any of the empirical methods that have been devised to deal with this problem. While tests on actual assembled coils cannot be regarded as a means of checking the absolute validity of these relations because that requires more fundamental work, it is true that the expressions derived from Lewis' and Merkel's relations are confirmed by test data. In this work the choice lies between accepting these fundamental relations which admittedly involve slight errors or that of assigning arbitrary characteristics to the surface.

Professor Tuve questions why we chose to deal directly with surface temperature rather than with the conditions of the refrigerant. In this connection it must be recognized that in cooling and dehumidifying two kinds of heat transfer are involved. From the air to the surface total heat is transferred as a function of total heat difference while from the surface to the water the same heat is transferred as a result of temperature difference. These two differences cannot logically be combined into a common potential without assigning to the total heat temperature line an arbitrary direction.

The choice of terminal surface temperatures as means of computing surface areas also involves the assumption made in the paper that the total heat temperature line may be taken as straight for limited ranges. This assumption leads directly to the mean heat difference used in the calculation of areas. Now despite the fact that the chord drawn between the end points on the total heat temperature involves

only a small error in relation to the absolute total heats for intermediate points for a range say 10 deg to 15 deg this is by no means the whole story. The error involved must be considered in relation to the potential. This means that a single step solution will not always provide sufficient accuracy in determining the mean heat difference and a two (or more) step solution is necessary as indicated in the text.

The use of terminal surface temperatures then makes it possible to determine initially from experience whether a single step is acceptable or whether a greater number is necessary. In a solution that does not investigate surface temperature this important factor might be ignored.

There was also a question regarding the use of enthalpy and employing wet-bulb as an index of this quantity. In this paper we employed Goodenough's tables and in the derivation of the surface temperature chart used wet-bulb as an index of these socalled total heat values. Actually wet-bulb is not exactly an index of either of these quantities but is so used because the error involved is not great in the range of air conditioning problems. To compute enthalpy it is necessary to know the dry-bulb and dew-point temperatures.

MR. DOWNS: You have referred to dry coils and wet coils, but of course, there are other applications of coils where you have frost and ice. Since the paper did not definitely limit the scope of this equation and your method, would you like to limit that scope not to include ice and frost, or would you extend it to that region?

MR. MCELGIN: In reply to Mr. Downs' question we have checked this theory against tests made under heavy condensation loads and have found good correlation. The theory is only applicable above the frost line and does not apply to frosted coils.

THE PERIPHERAL TYPE OF CIRCULATORY FAILURE IN EXPERIMENTAL HEAT EXHAUSTION

The Role of Posture

By Robert W. Keeton,* M.D., Ford K. Hick,** M.D., Nathaniel Glickman,*** M.S., and M. M. Montgomery,† M.D., Chicago, Ill.

IN a previous report [1] the effects of exposure of healthy human subjects to comfortable, hot dry, and hot wet conditions were noted. When the subjects were placed under the hot conditions there was a maximal dilatation of the peripheral vessels and an increase in circulating blood volume. If there was no rise in rectal temperature, the cardiac output remained unchanged. If the temperature rose above normal there was an increase in oxygen consumption and cardiac output. It is a matter of common knowledge that hot conditions establish in the individual a sensation of discomfort which ·becomes unbearable as the humidity rises. It is a matter of common experience that under such conditions either heat exhaustion or heat stroke with an associated hyperpyrexia may develop. The latter not uncommonly terminates in death. Evidently such an exposure exacts a strain on the circulation and should prove a useful weapon for the analysis of circulatory adjustments. It seemed wise, therefore, to study critically these adaptations of the circulation in the hot environment. The present report details such studies.

Methods

The subjects were all healthy medical students or nurses. They came in the experimental room in the evening and slept there. About 4:30 a.m. a floor

nurse awakened the subjects, who then threw off any covers they had been using and lay quietly on the bed until the observations were begun at about 7:30 a.m. Female subjects wore a costume of short trunks and a brassiere. The males were nude. All subjects were observed 12 hours after the last meal, and after a satisfactory rest. They were therefore under basal conditions. When the conditions of the experiment were such that no one could sleep in the room, the subjects slept in another room on the same floor, were awakened early and moved into the experimental room by 5 a.m. so as to be well adjusted to the environment.

The environments used in this study were approximately as follows: hot dry (99.5 F dry-bulb and 70.0 F wet-bulb), and hot wet (99.5 F dry-bulb and 90.0 F wet-bulb). As control studies the same procedure was followed in comfortable (85.0 F dry-bulb and 57.0 F wet-bulb) and in cool situations (75.5 F dry-bulb and 52.0 F wet-bulb). The air motion was minimal, probably less than 25 fpm at all times. The walls of the room are well insulated and contain no windows. Globe thermometer and thermal integrator readings have consistently shown the wall and air temperature the same, or very nearly so at all conditions.

RESULTS

Pulse Rate

A rise in pulse rate on exposure to heat is probably the best known of all circulatory adjustments to hot environments. Haldane [2] observed individuals in a hot room and in Turkish baths and noted that for every 1 F rise in rectal temperature there was an increase of 20 beats of the heart per minute. However, on returning to cool conditions there was an immediate drop in pulse. He concludes that the pulse rate depends not merely on the rectal but also on the external (wet-bulb) temperature.

Under conditions of light work in experiments on himself, Vernon [3] has noted that the relationship between pulse rate and body temperature is rather a complicated question. His experiments covered a rather wide range of temperature. From the individual records of his experiments, he concluded that the pulse rate is more dependent on the air temperature than on the body temperature. However, in a chart which shows the mean results of his experiments, there is a close correlation between body temperature and pulse rate.

McConnell and Houghten [4] observed subjects in the sitting position over a relatively wide range of environmental temperatures, and concluded that the correlation between pulse, wet-bulb, and humidity was significant. However, the correlation with effective temperature was slightly better.

Adolph and Fulton [5] also observed subjects in the sitting position, but exposed them "to conditions which cannot be endured by man for an indefinite length of time." In a saturated atmosphere at 105 F they concluded that

[2] The Influence of High Air Temperatures, by J. S. Haldane. (*Journal of Hygiene*, 5:494, 1905.)
[3] The Index of Comfort at High Atmospheric Temperatures, by H. M. Vernon. (*Medical Research Council Special Reports* No. 73, page 116, 1923.)
[4] ASHVE Research Report No. 654—Some Physiological Reactions to High Temperatures and Humidities, by W. J. McConnell and F. C. Houghten. (ASHVE Transactions, Vol. 29, 1923, p. 129.)
[5] ASHVE Research Report No. 672—Further Study of Physiological Reactions, by W. J. McConnell, F. C. Houghten and F. M. Phillips. (ASHVE Transactions, Vol. 29, 1923, p. 353.)
[5] Effects of Exposure to High Temperatures upon the Circulation in Man, by E. F. Adolph and W. B. Fulton. (*American Journal of Physiology*, 67:573, 1924.)

the heart rate is more nearly correlated with mouth temperature than with skin or rectal temperatures. When the subjects were withdrawn from the room and exposed to cool surroundings, the mouth temperature and pulse rate dropped correspondingly. The rectal temperature, however, lagged behind, remaining elevated for some time. They also noted from a restudy of McConnell and Houghten's data that similar conclusions could be reached.

A graphic correlation of the resting pulse rates with the rectal temperature is shown in Fig. 1 for this study.
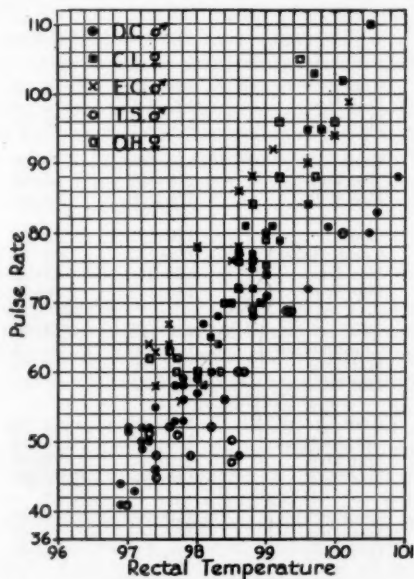
FIG. 1. PULSE RATES FROM FIVE SUBJECTS ARE SHOWN IN RELATION TO THE RECTAL TEMPERATURE. ALL SUBJECTS WERE LYING DOWN AND AT "BASAL" CONDITIONS

This experiment has certain advantages. It was conducted under much simpler and in some respects better controlled conditions than others previously reported. Air currents were not blowing on the subject and he was surrounded by an atmosphere which was at the same temperature as the walls. He had reached a steady state before either the pulse or rectal temperature was taken except in the hot wet conditions. In many of these experiments he had slept in the environment. In others he had been exposed to it for 3 hours. His energy expenditure was at the basal level since no work was performed or food taken.

Throughout all the experiments except those in the upper range, where

the rectal temperature rose above normal, there was no increase in oxygen consumption. Correlation between the rectal temperature and the pulse rate is good. Hence, under the conditions of the experiment one would have to conclude that the increase in the temperature of the blood in some way affects the pace make of the heart and accelerates its rate.

For the purposes of analysis one may divide the experiments cited in the literature into two phases, the experimental phase and the recovery phase.

TABLE 1. Arterial Blood Pressure of Five Subjects[a] (mm Hg)

Subject	Condition	Systolic	Diastolic	No. of Observations
C. L. Female	Cool	120–102	78–68	4
	Hot Dry	122–110	72–62	8
	Hot Wet	130–128	78–60	3
O. H. Female	Cool	114–106	72–68	4
	Hot Dry	118–100	68–58	8
	Hot Wet	120–106	82–64	3
J. W. Male	Cool	120–112	88–76	3
	Comfortable	118–112	92–78	3
	Hot Dry	128–116	84–70	5
	Hot Wet	136–122	80–58	5
W. C. Male	Cool	126– 98	72–60	3
	Comfortable	106–102	72–62	3
	Hot Dry	112–106	78–70	5
	Hot Wet	122–108	76–56	5
D. C. Male	Cool	108–100	72–60	4
	Comfortable	108–100	68–60	4
	Hot Dry	110–102	70–60	5
	Hot Wet	120–104	72–50	9
A. A. Male	Comfortable	110–106	68–68	2
	Hot Dry	112–104	72–58	4
	Hot Wet	110–92	58–50	2

[a] Subject in lying position.

From recent studies [6, 7, 8, 9, 10, 11] it is now appreciated that the body makes many adjustments directed towards the maintenance of body temperature. In the lower range of temperature heat loss by radiation and convection is important, but in the higher ranges evaporation is the determining factor. Hence, it is obvious that if one correlates pulse rate, which is an index of

[6] Physiological Reactions of the Human Body to Varying Environmental Temperatures, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge. (American Journal of Physiology, 120:1-1937.)
[7] The Mechanism of Heat Loss and Temperature Regulation, by E. F. Du Bois. (Lane Medical Lecture, Stanford University Press, 1937.)
[8] Life, Heat, and Altitude, by D. B. Dill. (Harvard University Press, Cambridge, 1938.)
[9] Role of the Extremities in the Dissipation of Heat, by W. G. Maddock and F. A. Coller. (American Journal of Physiology, Vol. 106, 589, 1933.)
[10] The Effect of Changes in Environmental Conditions on Skin Temperatures and the Dissipation of Heat from the Body, by C. Sheard, M. D. Williams and B. T. Horton. (Proceedings American Physiological Society, 1937, p. 147.)
[11] The Application of the Theory of Heat Flow to the Study of Energy Metabolism, by A. C. Burton. (Journal of Nutrition, 7:497, 1934.)

cardiac work, with factors in the environment, a factor should be selected operative through the entire temperature ranges.   The dry-bulb is the only factor so operative.   Investigators are agreed [12] that pulse rate and dry-bulb do not vary together.   The best correlation is with the wet-bulb within the ranges where it becomes effective in raising the body temperature.   Hence, there is good reason to accept the correlations of McConnell and Houghten [13] in the upper temperature ranges.   Adolph and Fulton [14] correlate the pulse rate with mouth temperature changes.   The data of Haldane [15] and of Vernon [16] also indicate a correlation with body temperature.   One must conclude, therefore, that under conditions of rising temperature, if a steady state can be obtained, the pulse rate will be determined by the body temperature.

Haldane, and Adolph and Fulton, have found difficulty in accepting this conclusion because of the sudden drop of the pulse in the cool recovery atmosphere while the rectal temperature remains elevated.   This presents no difficulty.   The pulse rate could not be expected to be controlled entirely by one factor.   In the recovery with exposure to cool air, evaporation is lessened, radiation becomes active, peripheral vaso-constriction sets in and the blood volume is shifted.   Hence, the pulse rate now becomes subjected to the control of different physiological mechanisms.

Arterial Blood Pressure

The arterial blood pressure of these resting subjects, taken while they were lying flat in bed is detailed in Table 1.   It should be realized that the readings were taken on different days.   Experimental conditions did not permit the continuous passage of a subject through all the environmental zones.   Hence, there are expected variations in the initial blood pressures.   There is not any great change in the observed pressures within the four environments.   A lowering of the diastolic pressures in the hot wet environment, where it was previously noted that a maximal dilatation of the peripheral vessels occurs, might be expected.   McConnell and Houghten,[17] observing subjects in the sitting position, found some variations in both systolic and diastolic blood pressures.   These changes showed the best correlations with the wet-bulb and effective temperature.   Adolph and Fulton report one subject (lying position) whose systolic pressure rose 10 mm and whose diastolic dropped to zero after 1 hour exposure to 105.3 F "supersaturated atmosphere."   No other data have been found on blood pressures of subjects in controlled hot environments in the lying position.   The observations, however, are not sufficiently numerous to be conclusive.   The results of this research would speak for the integrity of the mechanism delivering blood to the tissues in the hot environments under these conditions.

Venous Blood Pressure

With a constant blood volume, the venous pressure is responsible for the return of blood to the heart.   This pressure is a resultant of three factors: (a) the pressure transmitted from the arterial side through the capillaries,

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[12-17] Loc. Cit.   See Notes as follows: 1 and 4; 4; 5; 2; 3; and 4.

(b) the pressure developed from the tonus of the veins and the surrounding muscular tissues, and (c) the suction due to the negative pressure developed in the chest during inspiration. The positive venous pressure decreases from the periphery towards the heart. As the blood enters the chest the values fall to zero and become negative within the chest cavity. Hence, inevitably the venous pressures vary widely since factors (a) and (b) are not the same in two

TABLE 2. VENOUS BLOOD PRESSURE[a] (MM H_2O)

CONDITIONS	L.	H.	W.	CH.	CA.
Cool.............	113	96	140	101	128
	74	92	152	148	177
	122	99	119	...	...
	115	...	...	...	...
Comfortable........	...	...	109	125	138
	...	...	110	117	116
	...	...	109	101	99
	...	...	...	...	107
	...	...	...	...	104
	...	...	...	...	102
Hot Dry...........	74	73	96	91	82
	72	59	124	96	100
	73	57	102	90	102
	71	39	...	...	93
	79	53	...	...	91
	70	60	...	...	95
	59	...	...	...	84
	77	...	...	...	91
Hot Wet...........	63	40	126	72	87
	63	93	149	99	111
	90	25	129	106	90
	61	53	148	92	108
	...	...	107	105	107
	...	...	...	...	104
	...	...	...	...	101
	...	...	...	...	113
	...	...	...	...	110
	...	...	...	...	92

[a] Subject in lying position.

individuals, and may not be the same in a given individual under all circumstances. However, this pressure must be maintained above a critical level if the blood is to be returned to the heart. In measuring these pressures care must be taken to secure them at the same level with reference to the heart. In these experiments the venous pressure was measured by the direct method of Moritz and Tabora,[18] in which a good sized needle is passed into a large vein at the elbow and blood allowed to rise under its own pressure into an attached manometer tube which is already partly filled with 10 per cent sodium

[18] Ueber eine Methode beim Menschen den Druck in oberflächlichen Venen exakt zu bestimmen, F. Moritz and D. von Tabora. (*Deutsches Archive f. klinische Medizine*, 98:475, 1910.)

citrate solution. The latter prevents clotting in the needle and manometer tube. The pressure is then measured in millimeters of water and reduced to the level of the auricle of the heart. These data for 5 subjects in the lying position are given in Table 2. All the values fall within the accepted normal levels (40 to 110 mm of water). Therefore, the return of blood to the heart must be adequate in all environments observed with the subjects lying down.

Postural Adjustments

When the human subject assumes the upright position a considerable number of adjustments are necessitated if an undisturbed blood flow to all parts of the body is maintained. These adjustments have been studied in detail, especially by Heymans,[19] but a discussion of them is not indicated at this time. Suffice it to say that an individual may have an entirely adequate circulation in the lying position, but find that it is totally inadequate when he is suddenly placed upright and must make new hydrostatic adjustments. Difficulty in standing after exposure to heat has been described by Leonard Hill,[20] Adolph and Fulton [21] and Bazett.[22] More recently Nielsen, Herrington and Winslow [23] have noted that when their subjects were tilted to an angle of 70 deg they collapsed before routine observations could be completed. The observations previously reported have generally been incidental to other studies. Weiner [24] has reported that 6 out of 36 laborers who were able to work under hot wet conditions were afterwards embarrassed by the erect posture.

Using the same general routine as outlined here, the following observations were added to the list for each morning: The pulse rate and arterial blood pressure were recorded with the subject still lying down. Then the subject sat up on the side of the bed and the pulse rate and blood pressure were again taken. Then on standing these two items were recorded every 30 seconds until either an adjustment was made or until the subject felt faint and chose to lie down.

Six subjects have been studied. The general trend of the blood pressure records is the same in each of this small group. However, they differed somewhat in degree of capacity to tolerate heat. One woman fainted 4 times, one man became dizzy only twice.

The charts present the data from W. C., a medical student aged 22, which was typical. In the cool condition (Fig. 2) an initial slow pulse was associated with a low to normal pressure. On standing up the pulse rose to about 80, the blood pressure changed little and no symptoms appeared. In the comfortable conditions (Fig 3) the situation was much the same. The blood pressure varied in a moderate degree and the pulse rate rose slightly.

In hot dry conditions this subject (Fig. 4) did not run a fever and was all

[19] Le Sinus Carotidien, C. Heymans. (Bouckaert and Regniers, Doin, Paris, 1933.)

[20] The Influence of Hot Baths on Pulse Frequency, Blood Pressure, Body Temperature, Breathing Volume and Alveolar Tensions in Man, by Leonard Hill and M. Flock. (*Proceedings of the Physiological Society, Journal of Physiology* 38, p. LVII, 1909.)

[21] Loc. Cit. See Note 5.

[22] Studies on the Effects of Baths on Man. I. Relationship between the Effects Produced and the Temperature of the Bath, by H. C. Bazett. (*American Journal of Physiology*, 70:412, 1924.)

[23] The Effect of Posture on Peripheral Circulation, by M. Nielsen, L. P. Herrington and C.-E. A. Winslow. (*American Journal of Physiology*, 127:573, 1939.)

[24] An Experimental Study of Heat Collapse, by J. S. Weiner. (*Journal of Industrial Hygiene and Toxicology*, 20:389, 1938.)
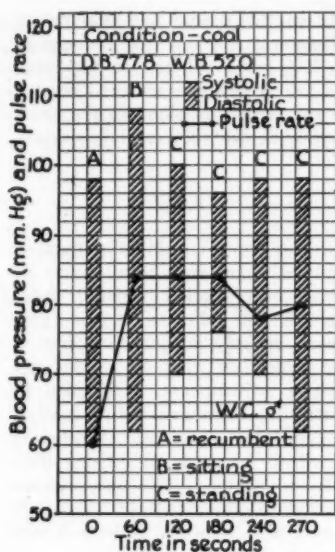
FIG. 2. IN A COOL ENVIRONMENT THERE WAS NO GREAT CHANGE IN BLOOD PRESSURE OR PULSE RATE ON STANDING. IN A FEW MINUTES THE BLOOD PRESSURE WAS IDENTICAL WITH THE INITIAL LEVEL
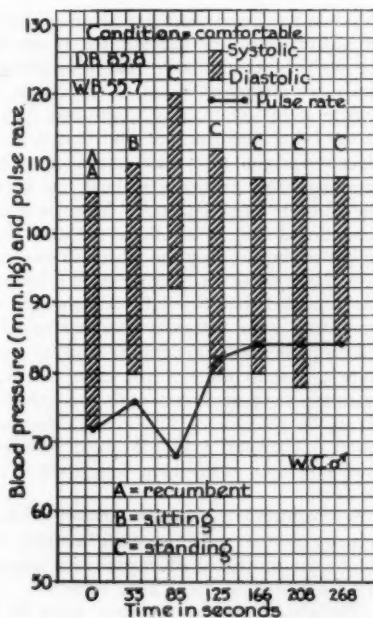
FIG. 3. IN THE COMFORTABLE ENVIRONMENT THE INITIAL RISE IN DIASTOLIC PRESSURE WAS SIGNIFICANT, BUT ADJUSTMENT WAS RAPID. PULSE RATE DID NOT EXCEED 84
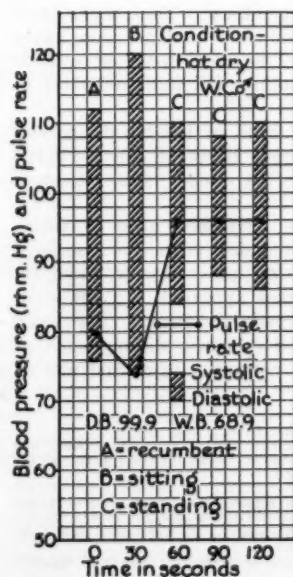
FIG. 4. IN HOT DRY CONDITIONS THE SAME SUBJECT SHOWED A SIGNIFICANT RISE IN DIASTOLIC PRESSURE, A LOWERED PULSE PRESSURE, AND A LEVELING OF THE PULSE RATE AT 96

right while lying down. He had a dry skin, or at times was wet with sweat on limited parts of the body. On standing there was a rise in pulse rate to above 100 and the blood pressure fell moderately, the pulse pressure narrowing somewhat. He was able to stand as long as several minutes with no complaints.

In contrast with the three situations just described, all of which were well
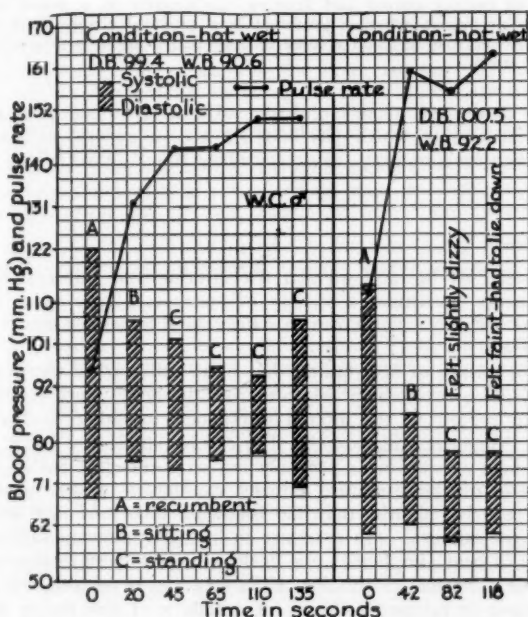
FIG. 5. ON THE LEFT WAS THE RESPONSE TO THE HOT WET CONDITION IN WHICH THE SUBJECT ADJUSTED SO THAT HE COULD REMAIN STANDING. HIS PULSE RATE THAT DAY REACHED 150 AND HIS BLOOD PRESSURE FELL TO 94 OVER 76, A PULSE PRESSURE OF 18. ON THE RIGHT, ON ANOTHER SLIGHTLY HOTTER DAY, THE PULSE WENT TO 164 AND, THE BLOOD PRESSURE FALLING QUITE LOW, THE SUBJECT HAD TO LIE DOWN

tolerated by the subject, come the findings in hot wet conditions (Fig. 5), with the subject having a fever. One graph presents compensatory adjustments; but in the other there is seen a progressive change until the subject chose to lie down. The blood pressure reached a level of 94 systolic and 78 diastolic, a pulse pressure of only 16, and still came back while the pulse rate was above 140. When he became dizzy and chose to lie down, his pulse was about 160 and the blood pressure was 76 mm systolic.

Such ideas about posture are not new. Leonard Hill [25] repeatedly mentioned the difficulty of maintaining an erect posture after a hot bath. Adolph and Fulton presented a theory of heat stroke as a peripheral circulatory failure. Bazett mentioned dizziness, etc., on standing after a hot bath. In a review Bazett [26] wrote the following:

". . . the maintenance of an erect posture, with the consequent tendency for blood to accumulate in the dependent lower limbs, may cause inadequate venous return to the heart, decreased cardiac output and fainting, particularly in a warm environment. Failure of cardiac output, whether produced by a postural change such as that just mentioned or by decrease in blood volume through dehydration, is generated by inadequate venous return."

Apparently those ideas about circulatory adjustment to both heat and posture are matters concerning which physiologists are in general agreement. The data from which such concepts, as described by Bazett, have grown are limited to date.

Heat Exhaustion

It is the belief of the authors that the fainting and faintness observed in these experiments constitutes a variety of syncope, a type of failure of the peripheral circulation.[27] There is also some reason to believe that it constitutes *heat exhaustion*.

Whereas the nature of heat cramps has been well worked out and shown to be the result of loss of sodium chloride from the body, there is no unanimity of opinion in regard to the altered physiology in either heat stroke with hyperpyrexia,[28] or heat exhaustion.[29] The careful studies by Dill [30] and others indicate that heat exhaustion is probably not on a chemical basis.

On the basis of the literature already mentioned in this paper, and on observations reported in this study the following theory of heat exhaustion is presented by the authors: On exposure to hot environments sufficient to lead to a retention of heat (i.e. fever), the healthy subject faces increasing difficulty in maintaining a normal circulation while standing or sitting. If sufficiently severe, faintness or fainting follows. A corollary of this theory is that subjects in poor condition, fatigued, or affected by disease, would find these circulatory adjustments more difficult, and would experience symptoms more readily than normals.

Conclusion

1. Circulation is well maintained in hot environments by healthy subjects while they are lying down, as is shown by normal arterial and venous pressures.

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[25] Loc. Cit. See Note 20.
[26] The Effect of Heat on the Blood Volume and Circulation, by H. C. Bazett. (*Journal American Medical Association*, 11, 1841, 1938.)
[27] The Nature of Circulatory Collapse Introduced by Sodium Nitrite, by Soma Weiss, R. Wilkins and F. Haynes. (*Journal of Clinical Investigation* 16, 73, 1937.)
[28] Heat Stroke, Clinical and Chemical Observations on 44 Cases, by E. B. Ferriss, M. A. Blankenhorn, H. W. Robinson and G. E. Cullen. (*Journal of Clinical Investigation*, 17:249, 1938.)
[29] Common Emergencies from Contact with Certain Physical Agents, by E. C. Elkins. (*Medical Clinics North America*, 22:1009, 1938.)
[30] The Ill Effects of Heat Upon Workmen, by J. H. Talbott, D. B. Dill, H. T. Edwards, E. H. Stumme and W. V. Consolazio. (*Journal of Industrial Hygiene and Toxicology*, 19:258, 1937.)

2. The adjustment of the circulation to the erect position becomes increasingly difficult for normals as the environment becomes hotter and fever appears.

3. Failure of this adjustment to the erect posture leads to symptoms of faintness or fainting, and constitutes, in the opinion of the authors, the clinical result known as heat exhaustion.

DISCUSSION

C.-E. A. WINSLOW: These results are of very great interest. Dr. Keeton's work is one of the few studies in which female subjects have been examined. Nearly all the work in this field, except a little by Dr. E. F. Du Bois has been done with young male subjects, and this represents a very important extension of the field.

It seems to me that the Society has a real interest in precisely fixing the upper limits of conditions which people can endure. I do not know that it is particularly interested in the details of heat stroke exhaustion, but it is very much interested in the determination of the maximum points, and the ways in which sex and individual susceptibility and posture and previous conditioning affect that response.

It seems to me that it ought to be possible to work out a borderline field of experimentation in this area of medical research which would be of continuing significance.

JOHN HOWATT: I had a little to do with the inception of this bit of research, and I am proud of it and proud of the way it is going. When it was started we were told that it was a 5-year project; that these researches in fundamentals cannot be completed in a few hours or a few days or a few months. I think this is the fourth year of the project. Dr. Keeton intimated that there was still further research to be done in it, and I believe it is worthwhile to carry it on to a conclusion where we can apply the knowledge that they obtain in the research.

I do want to compliment Dr. Keeton on the very excellent presentation, presented in such a way that I followed almost breathlessly. I think it was exceptional.

A. E. STACEY, JR.: This paper is of particular interest at this time because we have so many other investigations being carried on along these lines, and this paper goes beyond what we are doing in other places.

A question was just asked me, if Dr. Keeton has any indications of what happens to a person if he gets down to this exhaustion point and is brought out into proper conditioning. This may happen three or four times in perhaps that many days.

DR. F. K. HICK: The female subjects have been particularly tractible. We are able to hire nurses, for instance, in their post-graduate training who are accustomed to very low incomes for relatively low pay for this sort of experiment and find them very cooperative and not particularly susceptible to the emotional influences which might be an expected part of an experiment.

The question in regard to what happens when a subject is brought out of a hot environment into a cool environment—we have not paid sufficient attention to answer that. One of the female subjects has twice fainted after leaving the hot room and being in a cool room waiting for a shower. Some of the experimenters have had minor difficulties on leaving the hot room, but they are apparently unimportant and have not seriously affected their health.

These hot environments, of course, lead to a fever and a temperature of 100 or more has been observed in most of the experimental subjects. Apparently no serious

harm is entailed in repeated fainting. We have had one girl who fainted at least five times and who is quite willing to go back in for more experimental work if we wish. Apparently no real harm is done to healthy subjects in that circumstance.

To my way of thinking, the importance of this work in regard to industry should not be minimized. So often in plants, steel mills, for instance, where as a medical student I had a chance to make a living, workers around hot rollers topple over frequently and generally spend three or four days or more off work after such a fainting episode.

There is considerable mystery to the mechanism of these heat exhaustion states in the minds of a good many people engaged in industrial work. As a matter of fact, they seem relatively simple and as long as the patient can lie down while he is in such an exhaustion situation he will recover at once. By the time he is in a company hospital he will apparently be a nearly normal individual except for fatigue.

I think that is not a matter of common knowledge at the present time.

W. R. RHOTON: I would like to know if, in the high dry tests, the relative humidity together with increasing the air motion overcomes the effects on the subject, or have you made experiments on this basis?

DR. HICK: Experiments have all been conducted in still air for the purposes of simplicity of control. Logical extension of the work would be in the direction you indicate.

ADVANTAGES OF BACTERICIDAL ULTRA-VIOLET RADIATION IN AIR CONDITIONING SYSTEMS

By Harvey C. Rentschler * and Rudolph Nagy,** Bloomfield, N. J.

I T is the function of modern air conditioning systems to provide rooms with air that is at the proper temperature and humidity and is practically free from dust and dirt. This is generally accomplished by recirculating the air from the room with the addition of a small amount of fresh air through the necessary heating or refrigeration equipment designed for this purpose. The air in occupied rooms, at times, is highly contaminated with bacteria. Wells[1] demonstrated that bacteria floating in the air may remain in an active state for a considerable time and that air-borne pathogenic bacteria may be responsible for the transmission of certain respiratory diseases.

Ultraviolet radiation of certain wave lengths has long been known to have bactericidal properties and has been proposed as a suitable agent for killing air-borne organisms. There are three ways such radiation can be applied.

1. *By exposing the air in the room to the direct radiation from an ultra-violet source:* This method is suitable for special cases only, as for example, in hospital operating rooms as shown in Fig. 1 or for laboratory transfer rooms. In such places it is possible for the occupants to protect themselves against excessive exposure from direct radiation.

2. *By the use of semi-direct radiation:* The air in part of the room not directly occupied is irradiated while other parts are shaded from the direct rays. This type of irradiation is particularly suitable for such places as nurseries. A typical arrangement of ultraviolet radiation lamps in a small hospital nursery room is shown in Fig. 2 and Figs. 3 and 4 are photographs of such an installation.

3. *By placing the ultraviolet lamps in the ducts of air conditioning systems:* In this manner all the recirculating air is exposed to the radiation as it approaches and passes the lamps in the duct, an installation of which is shown in Fig. 5.

The work to be described here was undertaken to determine the practicability of adding bactericidal ultraviolet radiation to air conditioning systems. In a recirculating system provided with bactericidal ultraviolet radiation the air

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[1] Air-Borne Infection, by W. F. Wells and M. W. Wells. (*Journal American Medical Association*, 107, 1698 and 1805, 1936.)
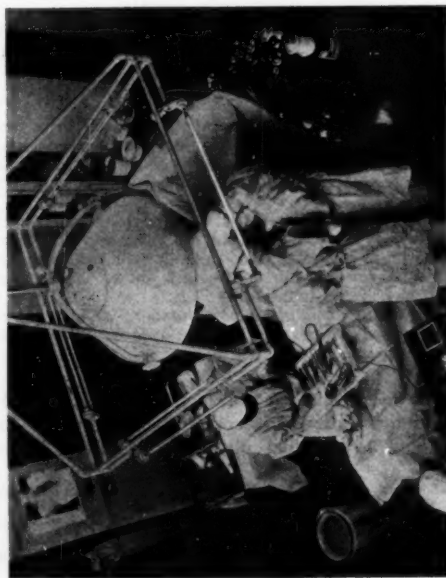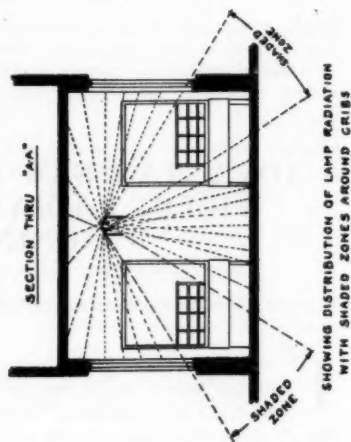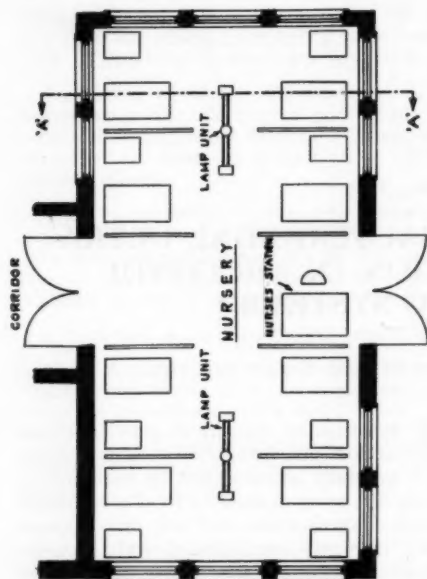
Fig. 1.   Ultraviolet Lamps Over Operating Table at Duke Hospital, Durham, N. C.

Fig. 2.   (Right) Shrouded Lamp Units in Infant or New Born Nursery

would not only have the desired temperature, humidity and cleanliness but would be reasonably free from bacteria as well.

Before an intelligent use of radiation can be made it is highly desirable that the nature of the killing of bacteria by ultraviolet radiation be understood and essential that the factors controlling this action be fully considered.

Hundreds of tests were conducted on various organisms seeded on Petri plates in order to obtain some relative value of their resistivity to ultraviolet radiation. Table 1 will indicate such values. These values were substantiated at various university laboratories especially on those organisms of pathogenic nature. Escherichia coli was selected as an index organism primarily because it is non-pathogenic and because of the constancy of this organism under

TABLE 1. RESISTIVITY OF VARIOUS ORGANISMS TO ULTRAVIOLET RADIATION

MICROORGANISM	CLICKS ON STANDARD TANTALUM CELL METER
BACTERIA	95–100 PER CENT KILL
E. Coli.	25–30
Staph. Aureus.	24–30
Staph. Albus.	25–30
Strep. Hemolyticus (Alpha Type).	22–25
Strep. Hemolyticus (Beta Type).	28–35
Shigella Flexneri.	25–30
E. Typhosa.	30–35
B. Anthracis.	50–65
C. Diphtheriae.	40–50
S. Marcescens.	25–30
N. Catarrhalis.	24–35

varying conditions and further because its resistivity is of the same order of magnitude as most of the pathogenic organisms usually associated with respiratory diseases. Values given are arbitrary and were obtained by the use of a special integrating ultraviolet meter.

Method of Measuring Ultraviolet Radiation

The integrated amount of ultraviolet radiation to which the bacteria on a plate were exposed was measured with a tantalum photoelectric cell placed at the same distance from the ultraviolet source as the surface of the seeded plate. Such a cell constructed in the proper ultraviolet transmitting glass has a wave length response very similar to the bactericidal action of the radiation from about 3000A to approximately 2300A. By using an ultraviolet source in the same glass the shorter wave length radiations are absorbed by the glass container and the cell response and the bactericidal action for different wave lengths agree as closely as any tests that can be made.

To measure the integrated amount of exposure, a battery B charges a condenser C proportional to the effective radiation falling on the cell P, Fig. 6. When the condenser is charged to the breakdown potential from a to b of the glow tube G, it discharges from a to b, thereby igniting the glow from
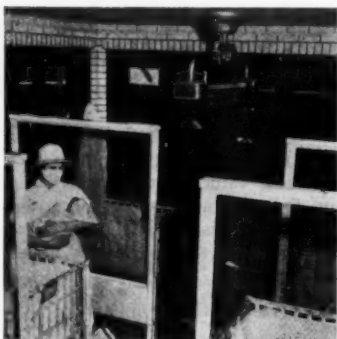
Fig. 3. Semi-direct Ultraviolet Lamp Fixture Installed in a New York Hospital Nursery

Fig. 4. Same Unit as Fig. 3, Used as a Sterilizing Agent for the Cribs with Infants Removed
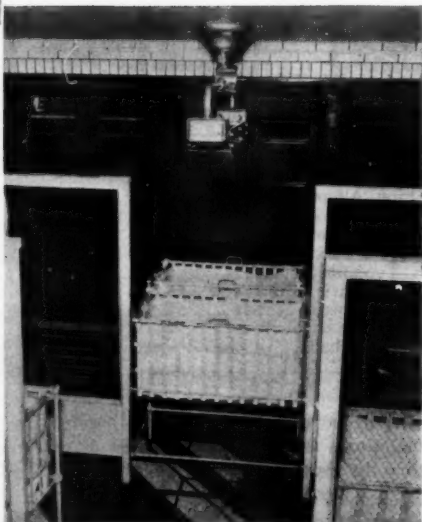
Fig. 5. Radiation Chamber in Air Duct of Auditorium Air Conditioning System
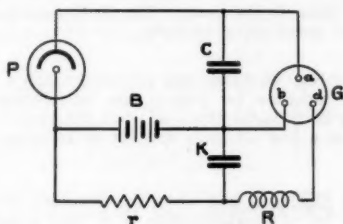
Fig. 6. Ultraviolet Meter Circuit

d to *b,* discharging the big condenser *K* through the electromagnetic counter *R*, thereby recording the number of times the condenser was charged by the radiation in question.

Factors Controlling Destruction of Bacteria

It is beyond the scope of this article to describe in detail the experiments conducted in establishing the principles of this application. These are published elsewhere.† However the following are briefly a few of the factors which are important in the use of such radiation for the destruction of bacteria in ducts of air conditioning systems.

1. *The bactericidal action is determined by the amount of radiation to which the bacterium is exposed regardless of whether a high intensity is applied for a short time or a low intensity for a correspondingly long time provided that the product of*

Exposure times:

1	2	3	4
10 min	40 min	1 hour 33 min	2 hours 43 min
5	6	7	8
4 hours 18 min	6 hours 20 min	8 hours 23 min	10 hours 58 min

Fig. 7. Seeded Petri Plates After Exposing Upper Halves to Same Total Amount of Radiation in Progressively Lower Intensities With Correspondingly Longer Times

† Bactericidal Effects of Ultraviolet Radiation, by H. C. Rentschler, Rudolph Nagy and Galina Mouroniseff. (*Journal of Bacteriology*, 41, 745-774, 1941.)

the intensity times the time is the same: This is known as the reciprocity law as applied to the action of radiation on bacteria.

To prove this, a condenser discharge was passed through a lamp to produce a high intensity of ultraviolet radiation for only a few microseconds duration. This was found to have the same bactericidal effect as when the same total amount of effective radiation was applied at a low intensity and for as much as several hours at room temperature.

From this it may be concluded that when the exposure of bacteria to radiation in a duct will necessarily be of short duration it is both practical and economical to provide sufficient intensity to obtain the total quantity of radiation required for their destruction.
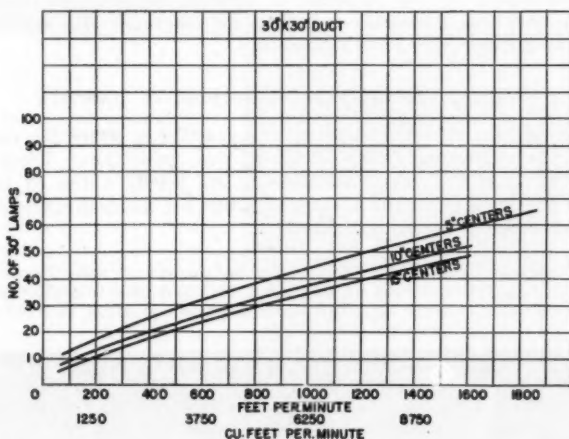
FIG. 8. CURVES SHOWING THE NUMBER OF LAMPS REQUIRED FOR VARYING VOLUMES OF AIR FOR A 30 IN. x 30 IN. DUCT

2. *The lethal action of ultraviolet radiation is not influenced by the temperature of the bacteria at the time of exposure:* A number of Petri plates were seeded with E. coli by a special method so that when not exposed to radiation all of the plates developed approximately the same number of colonies after incubation. A few such plates were incubated without exposure for use as controls. Half of the remainder were then placed in an incubator at 37 C for half an hour and exposed at the same temperature to a definite measured amount of ultraviolet. The rest of the plates were kept in a refrigerator at about 5 C for half an hour and exposed while cold to the same amount of radiation. The percentage of the colonies killed in the two tests was the same within experimental error.

3. *The resistivity of a bacterium to ultraviolet radiation varies appreciably at different stages of its life cycle:* This is indicated in the photograph Fig. 7. A number of Petri plates were seeded alike with E. coli. The upper halves of the plates were exposed to the same amount of radiation but in progressively lower intensities and with correspondingly longer times. This test is an extension of the experiment described herein under reciprocity law only here the bacteria were exposed while passing through one or more life cycles. This increased lethal effect does not signify

a breakdown of the reciprocity law but rather a change in the resistivity of the organism at different stages of its life cycle. This test shown in Fig. 7 was conducted at 32 C.

4. *Air-borne bacteria require the same amount of radiation as is necessary to kill the same organisms seeded on the surface of a Petri plate:* This was proved by a long series of experiments so designed that the distribution of bacteria at the different stages of their life cycle was the same at the time of exposure for the air-borne and for those seeded on the surface of the Petri plate.

5. *Air-borne bacteria at high humidity require the same amount of ultraviolet radiation for equal destruction as do the same organisms floating in low humidity air:*
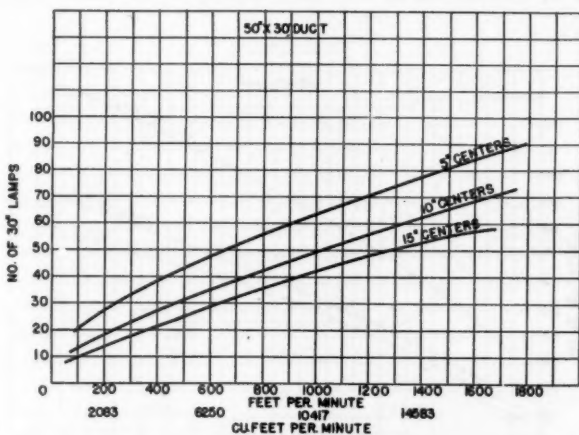
FIG. 9. CURVES SHOWING THE NUMBER OF LAMPS REQUIRED FOR VARYING VOLUMES OF AIR FOR A 50 IN. × 30 IN. DUCT

This is extremely important for the proper application of radiation for air sterilization in ducts and in high humidity food storage rooms.

6. *In a recirculating system the lethal action of direct radiation is the dominating bactericidal agent:* Ozone which may incidentally be produced by the lamps has an insignificant effect in comparison with the action produced by the radiation itself.

LAMPS AND LAMP CHARACTERISTICS

The radiation from the ultraviolet lamps [2] used in these tests is produced by an electrical discharge between cold electrodes through a mixture of inert gases and mercury vapor in a slender tube of special ultraviolet transmitting glass. These lamps have a high efficiency in producing the desired wave lengths for bactericidal purposes without generating a deleterious amount of ozone. They operate only a few degrees above ambient temperature and, therefore, heat correction in any application is unnecessary. They are commercially available in 10, 20 and 30-in. lengths. The lamps may be operated

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[2] Data and observations reported in this paper are based on tests conducted with an ultraviolet source known by the trade mark Sterilamp.

singly or as many as four in series on a special transformer. They are reliable in operation, require about 15 watts each, and have a practical operating life of about 4500 hours continuous burning. The lamps produce about 80 per cent of the radiation at or near the most effective bactericidal wave length in the ultraviolet spectrum.

Lamps in Air Conditioning Ducts

The proper installation of the lamps in air conditioning ducts requires considerable care. A definite amount of radiation is necessary to kill a given percentage of the bacteria as they move rapidly in the duct. The intensity
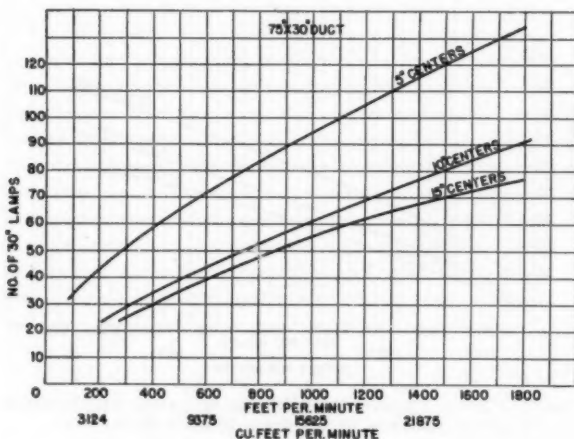
Fig. 10. Curves Showing the Number of Lamps Required for Varying Volumes of Air for a 75 in. x 30 in. Duct

of the radiation is greatest at the surface of the lamp and falls off with increasing distance. The most economical use of radiation in a duct, therefore, is under conditions in which there is nearly a uniform intensity through the cross-section. Lamps should be installed in a straight section of the duct in order to utilize the radiation on the bacteria before they reach the lamps and after they pass. While the effect of radiation on organisms diminishes with distance nevertheless the effect is cumulative. In many cases a long straight length is not always available; however, the minimum straight portion of the duct should be at least twice the length occupied by the lamps. In order to arrive at the amount of radiation that must be provided in a given duct the problem is first to determine the intensity at different distances from one lamp and then to find the intensity at any point in an array of uniformly spaced lamps. From these data it is possible to determine an average intensity of radiation along the duct and from this, the length of duct which must be

irradiated in order that sufficient energy is supplied to kill bacteria as they pass through the duct. In Fig. 8 the relation between the number of lamps required to produce satisfactory bactericidal action on E. coli and the different volumes of air passing through the duct per minute for a 30 x 30 in. radiation chamber is shown. Fig. 9 gives a similar relation for a 50 x 30 in. chamber and Fig. 10 for a 75 x 30 in. chamber for 5, 10, and 15-in. centers between lamps. As a practical illustration, an air conditioning system with lamps in the duct was installed in an auditorium which was designed to circulate 5800 cfm. Using a 75 x 30 in. duct for the radiation chamber, the minimum number of lamps required for this volume of air and for satisfactory action
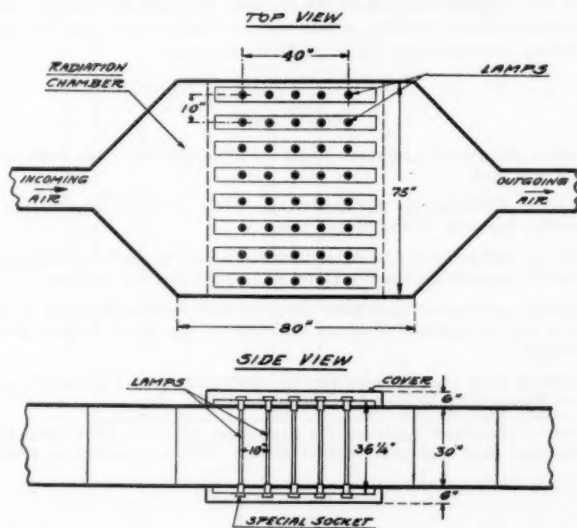
Fig. 11.   Schematic Arrangement of Lamps in Duct of Auditorium Air Conditioning System

on E. coli is obtained from the curves of Fig. 10. For 5 in. spacing this number is 57, for 10 in. spacing 33, and for 15 in. spacing 29. Since four lamps can be operated on the same transformer it was decided for experimentation to install 40 lamps in a chamber 75 x 30 in. on 10-in. centers— that is, 5 rows of 8 lamps per row and a lighted length of 40 in. or a total length of 80 in. of treatment duct as illustrated in Fig. 11.

Tests of the Installation

A humidifier was used to spray E. coli into the air of the auditorium. The spray mixture was made by diluting 5 cc of a 24-hour culture of E. coli with 50 cc of sterile broth and 2 liters of water. This mixture was sprayed near the air intake duct using only recirculated air. A Wells centrifuge was used

to test the effectiveness of the lamps in the duct by taking samples of air at the outlet grilles entering the auditorium. During these tests the filters were removed so as to obtain the effectiveness from the radiation alone.

Hundreds of tests were conducted to prove the efficacy of this installation under varying temperatures, under varying humidities and with varying amounts of radiation in the duct. The procedure in each test was to take samples of air with the lamps *off*, followed by samples with the lamps *on*, and again with the lamps *off*. From a comparison of the average of the two controls, namely, the first and third, and the sample of the air with the lamps *on* the percentage killed by the radiation was computed. The 40 lamps gave an average reduction of 99.5 per cent while 32 lamps gave a reduction of 98.2 per cent in the number of organisms passing through the duct. Neither the dehumidifier coils of the cooling system nor the filters were found to have an appreciable effect in removing bacteria from the air.

Conclusions

1. The use of ultraviolet radiation in air conditioning ducts has been found to be commercially practical.

2. There is no difference in the bactericidal effect of ultraviolet over a range of relative humidity between 35 and 95 per cent.

3. There is no difference in the bactericidal effect of ultraviolet radiation over the normal range of temperature usually found in air conditioning systems.

4. Calculations and curves have been prepared and checked by tests on a practical installation so that the number of lamps required for any given volume of air can be readily estimated.

5. For bacteria with greater resistivity to ultraviolet radiation than E. coli the intensity of radiation must be increased proportionately.

6. The use of ultraviolet radiation for destroying bacteria, when properly applied, has the advantage over other bactericides in that nothing is added or removed from the air.

References

1. An Ultra Violet Light Meter, by H. C. Rentschler (*Transactions, American Institute of Electrical Engineers*, 49, 576: 1930).

2. The Bactericidal Effect of Very High and Low Intensities of Ultra Violet Radiation, by H. C. Rentschler and R. Nagy (Report presented at *American Association for the Advancement of Science* Meeting at Richmond, Va., December, 1938).

DISCUSSION

W. F. Wells (Written): Agreement upon principles applied to the bactericidal irradiation of air should precede a critical examination of design. Dr. Rentschler propounds six *factors controlling the destruction of bacteria*. Four of these factors which relate the lethal action of ultraviolet radiation upon bacteria exposed on agar plates and in liquids are in general accord with the mass of work done on this subject. The fourth and fifth factors, however, which treat the question of the bactericidal action upon microorganisms suspended in air are at variance with other work on this subject.

It was stated by Wells and Fair in 1935 that:

It should be apparent from these tests that the destruction power of ultraviolet light of itself is of a higher order of magnitude in air than in liquids or other environments not highly transparent to ultraviolet radiation.[3]

Wells stated later that year:

Humidity is an important factor limiting the germicidal effect of ultraviolet light; an increase of 25 grains of moisture per pound of dry air may cause a loss of almost 90 per cent in bactericidal power. Whether this is due to absorption of the active wave-lengths or to other factors protecting the bacteria from the light has nòt yet been determined.[4]

These results are illustrated in the paper referred to by Dr. Rentschler.[5]

During the following year Dr. B. A. Whisler[6] conducted an exhaustive research on the quantitative relationships between exposure and mortality of E. coli suspended
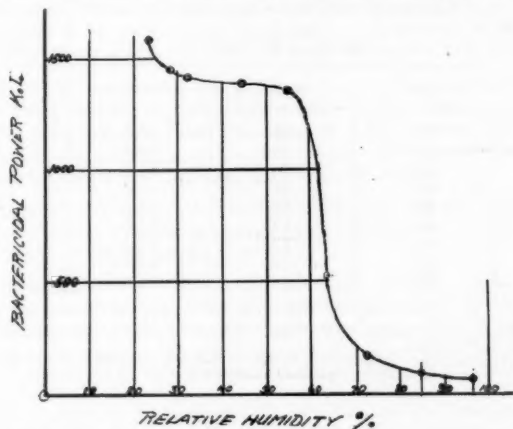
FIG. A. RESULTS OF 54 RUNS REPRESENTING OVER 500
AIR SAMPLES

in air of different humidities. The results of 54 runs representing over 500 air samples, computed from tests on a quartz Uviarc operating at 70 volts D.C., are retabulated and replotted on Table A and Fig. A. The bactericidal effect of ultraviolet light upon E. coli suspended in air obviously drops abruptly below a tenth of its former magnitude within a critical region between 50 and 75 per cent saturation. A less exhaustive study of a low pressure quartz tube of the Cooper-Hewitt type gave similar results, though difficulty in controlling arc voltage under the experimental conditions did not permit the clear-cut determinations shown for the Uviarc. Preliminary experiments upon E. coli suspended in flowing water and air under identical conditions further confirm this phenomenon, and the results have been corroborated by the work of Dr. L. R. Koller.[7]

[3] Viability of B. Coli Exposed to Ultra-violet Radiation in Air, by W. F. Wells and G. M. Fair. (Science, 82:280, September 20, 1935.)
[4] Air-Borne Infection and Sanitary Air Control, by W. F. Wells. (Journal of Industrial Hygiene, 17:253, November, 1935.)
[5] Loc. Cit. Note 1.
[6] The Efficacy of Ultra-violet Light Sources in Killing Bacteria Suspended in Air, by B. A. Whisler. (Journal of Science, Iowa State College, 14:215, April, 1940.)
[7] Bactericidal Effects of Ultraviolet Radiation Produced by Low Pressure Mercury Vapor Lamps, L. R. Koller. (Journal of Applied Physics, 10:624, September, 1939.)

In view of the mass of evidence indicating the protective effect of humidity, it is difficult to accept Dr. Rentschler's conclusion on this point without further data which are not presented in his contribution.

Hygienic specifications for the bactericidal irradiation of breathing air directly apply to the air returning to the circulating system and only indirectly to the air leaving that system. The bacterial concentration of the air breathed in the room depends upon an equilibrium determined by percentage of recirculation.[s] Thus, with 10 per cent recirculation, the increase in the rate at which organisms are vented from the room with 99.5 per cent reduction in the duct over that resulting from

Table A—Effect of Relative Humidity Upon Bactericidal Power[a] of Uviarc

(Compiled from B. A. Whisler[b])

Number of Runs	Relative Humidity Per Cent			Bactericidal Power K_0l_0
	Low	High	Average	Average
10	22	24	23.0	1588
7	27	29	28.0	1460
3	30	35	31.7	1420
4	41	48	44.0	1395
9	50	58	54.3	1363
7	60	67	63.0	530.3
4	71	73	71.8	174.2
8	81	89	84.1	95.1
2	95	97	96.0	69.0

[a] Bactericidal Irradiation of Air, by W. F. Wells. (Part I, Physical Factors, *Journal of the Franklin Institute*, 229:347, March, 1940.)
[b] The Efficacy of Ultra-violet Light Sources in Killing Bacteria Suspended in Air, by B. A. Whisler. (*Journal of Science*, Iowa State College, 14:215, April, 1940.)

98.2 per cent reduction in the duct is less than a tenth air change per hour— certainly a small gain for the difference in the number of lamps required: 40 as against 32. Twenty lamps would have substantially the same ventilating effect upon the air in the room.

The law of diminishing returns thus sets in early for irradiation in air-conditioning systems.

John Howatt: We are used to considering black light or ultraviolet as all of the solar energy that has a wave length less than $\frac{39}{100}$ of a micron and a rate of vibration greater than 750 billion per second. You made the statement that only a part of what we call ultraviolet is effective as a germicidal agent. As a point of information, I would like to know the limits of wave length and frequency of vibration that are effective.

H. C. Rentschler: The wave lengths that are effective in killing bacteria have been pretty well established by Gates and Hollaender and others. Very little bactericidal action is obtained with wave lengths longer than 3,000 Ångstroms, that is 300 millimicrons. As the wave length is decreased the effectiveness increases to about 2,650 or 2,660 Ångstrom units and then the effectiveness decreases again down to about 2,375 Ångstroms, after which there is an apparent upturn in the curve again.

[s] Measurement of Sanitary Ventilation, by W. F. Wells and M. W. Wells. (*American Journal of Public Health*, 28:343, March, 1938.)

So that for effective bactericidal radiation you must be below 3,000 Angstroms and the reason for this discharge through a mixture of inert gas and mercury vapor is that practically all the radiation is concentrated at or near the peak of the effectiveness; that is the resonance radiation from mercury 2,537, which is nearer to the peak of the most effective radiation than any other source of radiation known.

So that with the least amount of energy the greatest amount of bactericidal radiation is supplied.

MR. HOWATT: That would be $\frac{25}{100}$ of a micron wave length.

C. F. EVELETH: In describing your system, you spoke of it as a recirculating system. Do I understand you took no outside air at all, but merely the air of the room?

MR. RENTSCHLER: In some of the tests we only recirculated the air from the room and added to that room a large number of bacteria, so that we would have enough to get a good representative count of the number of bacteria that were killed.

In other tests we took nothing but air from the outside, particularly on very dry days, and in other tests on very humid days when it was raining, so we could have the high relative humidity to verify by means of a duct system the result that we had established by other methods which were far more accurate and far more delicate in determining any effect due to increased relative humidity.

Of course, in a normal system you could add some of the outside air to the recirculated, but what we were interested in was simply the amount of killing of bacteria by means of the radiation.

C.-E. A. WINSLOW: I think every year evidence is added to show the importance of this problem, evidence not only of a laboratory nature but definite evidence from clinical experience particularly in English hospitals. This is something which I think is going to be more and more vital in the whole field of air conditioning, and I have no reason to dissent with any of the general conclusions advanced. I do want to say, however, that while it is impossible without the data to criticize the bacteriological techniques, in a number of respects they run completely counter to all the work that everybody else has done and to basic bacteriological knowledge. Every one else has found that the effect of bacteria on air and on plates was very different, and it would be hard for a bacteriologist to conceive how it could help being so, how the bacteria on the plate could possibly help being somewhat protected.

As to the effect of relative humidity, that is based on very extensive and careful researches by Professor Wells who has been a leader in developing this whole business.

As to this varying susceptibility of bacteria at different phases of the process of cell division, if there is any such thing, that is something for which bacteriologists have been looking for 50 years without finding any slightest inkling that any such variation in susceptibility takes place as the cells divide.

J. J. AEBERLY: I think that we could simplify this whole problem if we could strip it of its living aspect and discuss it strictly from a chemical standpoint. The process of destroying bacteria, as given in this paper, is essentially the use of energy in the form of ultraviolet rays to cause a chemical change of the amino acids which are contained in all proteins. It appears to me that if we can agree to this premise Dr. Rentschler could answer more directly our questions and enable us to clear up our uncertainties. For a knowledge of the quantity of ultraviolet rays which will be necessary to complete the photo-chemical action, we should know the amount of amino acid present.

I cannot agree with some of the findings in this paper. It would seem that variation in dust content in the air stream may prevent this energy from reaching the bacteria to be destroyed; also that some of the bacteria similar to the spore-forming organisms would offer an additional complex problem.

W. L. FLEISHER: This is a subject that has interested all of us and has interested us particularly who have been involved in air filtration. I would not raise the point at all if the emphasis on its use in air conditioning had not been made so prominent.

I am interested in Dr. Rentschler's explanation of the size of the particle on which his bacteria must necessarily be carried in order to preserve its life. Of course, the bacteria which we saw under the microscope were all probably in a drop of water in which they would readily live and be preserved, but our air-borne bacteria, our coli or mole spores undoubtedly are carried on particles of a very definite size.

Now, great emphasis has been placed on the utilization of this method in recirculating ducts. If the particle sizes on which these bacteria are carried are sufficiently large (and in our conception this bacteria-carrying particle is of a size which is well above the 1 micron diameter), they can definitely be removed in wet processes.

When I was in England with Dr. Thomas Bedford he explained to me that in a thoroughly-sealed room, a room sealed as tightly as they could seal it, under the auspices of the Department of Tropical Diseases, they found that there was 20 per cent leakage into the room under any circumstances and consequently as all of the air which enters the room carrying particles with air-borne microscopic bacteria on it can readily go through the leakage sections of the room, you do not really cover all of the bacteria that can come into your room in your recirculated ducts.

The emphasis that I think could possibly be placed on these investigations would cover isolated portions of the room; that is, the efficiency of a method of this kind possibly over the operating table; but to emphasize its necessity in a return duct or a recirculating duct, where your air passes rapidly, in which all of the air in the room probably does not pass and in which the time element is a consideration, I think is putting emphasis on the wrong side of this particular phase of air cleaning.

In experiments that I carried on myself with very eminent bacteriologists in a wet method of cleaning air introduced from outside, we found on incubation, although the Petrie dishes showed a very large colony of various kinds of coli and mole spores, although we were positive that our cleaning method did not remove anything below 1 micron in diameter, we had no incubation of coli of micro-organisms beyond the cleaning apparatus, although in the water chamber itself we indicated a vast number of these micro-organisms which were readily destroyed either by ultraviolet rays or by other methods.

Consequently I am interested in whether the Doctor's investigations covered the sizes of the particles carrying his organisms and whether, if they are above a certain particular size, it is essential to go to this type of destruction or removal by the method he has indicated.

I am always a little bit bothered about making a very interesting and a very necessary experiment too embracing.

H. M. HART: In these experiments the life of these lamps was given, I think, at 4500 hours. Is there any difference in the effectiveness of these lamps over that period of their life? Are these tests made with new lamps or lamps that have been used for a period of hours, and what is the difference between the new lamp and the lamp approaching the end of its life?

MR. RENTSCHLER: New lamps, of course, are more effective than old lamps. There is a certain decrease in output with age of the lamp, but when we say the life is

4500 hours we mean the effective useful life. New lamps have more than the necessary amount of radiation that is required, according to these curves that I have shown, but we aim that the average lamp at the end of 4500 hours should still have sufficient effective radiation to produce the effect that we claim.

Reference was made to Professor Wells' work. When we could not get his results on humidity, I took special pains to get the article from which Professor Wells' conclusions are taken. This article is an unpublished thesis which is in the Harvard Library, and for the purpose of our discussion here, for summing up the field, I am just going to quote from this article:

"The fundamental assumptions used in this analysis of the action of ultraviolet upon bacteria are as follows: The Quantum Theory may be applied to biological killing." Then two other assumptions which are not of particular importance here, and lastly: "all organisms of one species are identical."

We have a vast amount of evidence that all organisms of one species are not identical, but that it does require a very different amount of radiation to kill different organisms depending upon what stage of the life cycle they are in.

Another point that was applied to this unpublished thesis is the difference between our work and the work that was performed at Harvard. In our work, in every experiment made, we measured the amount of radiation. In the work that was done at Harvard—now I am not trying to criticize the work; it was a beautiful piece of work; the work was well done at that time; a suitable ultraviolet measuring device was not available at that time; but in the conclusions drawn I am going to quote what they say: "A similar humidity series was secured by using a 14-in. low pressure mercury arc," where before they had used the regular mercury arc. "It will be seen that the irregularities are much greater than those observed with the uviarc. Although corrections were made for variations in voltage, the difficulties of measuring this variable with sufficient accuracy, coupled with the fact that the low-pressure lamp is sensitive to small changes in operating characteristics may account for a large part of these variations."

In the conclusion towards the end of the paper they say, "It is believed that the method of investigation followed during the latter part of the present study could be carried much further as a means of making an accurate statistical study of the reaction of bacteria to ultraviolet radiation. This will require, however, that provisions be made for careful and accurate comparison of the light source, as well as for the continuous measurement of output at individual wave lengths; a far more accurate explanation of the observed humidity effect might be made if this was done." And that is what we claim to have done.

With all of the data that we have accumulated I cannot escape the conclusion that different bacteria in the same culture and of the same species have a different resistivity and this must be considered in arriving at a conclusion as to the effect of humidity on the bactericidal action of ultraviolet radiation.

In answer to your question, I quite agree with what you say, but I want to call attention to one fact. Suppose I am the source of a contamination by coughing. I sit right next to you and I cough right into your face. No ultraviolet light will take out those organisms to be destroyed by the ultraviolet light.

So that the place of contamination in a room makes it doubly difficult to get the data that you are asking for, and all that we claim is this: we know that if you start a source of contamination in a room, say for instance in this room, way back in the corner and we put our testing device up here, in a relatively short time we can pick up the bacteria in this portion of the room.

If we put in a recirculating system, we also know that they will get up here much quicker than they will otherwise. We cannot avoid a certain amount of that transfer from one place to the other, but what we can do very definitely is in a recirculating system to avoid their being picked up in one place and brought into another place in a very short time.

I had occasion to test an air conditioning system for a dentist who had a number of small units. The air was all taken through the same circulating system. We placed the contaminator in one room and in a very short time we were able to pick up the bacteria in every one of the other rooms. If the air conditioning system had not been in, that contamination would have been very largely, if not entirely, confined to that one room, and that is why we claim, or we believe, it is at least a practical starting point to put the radiation in the duct and as time goes on, as we learn more about the way of handling the problem, to also put a certain amount of radiation in the room to take care of some of these other factors.

As far as the last point that you brought up goes, namely: did we take into account the different size of the particles? Yes. In the vast number of experiments that we made we investigated large size droplets, and small size droplets. We tried to expose them at high humidities and at low humidities, on plates and floating in the air. I am convinced that the droplets are sufficiently transparent so that the bactericidal action is not determined by the size of the droplet in which the organism is floating.

In the written discussion Wells points out that the radiation required for destroying air-borne bacteria and the effects due to relative humidity reported in this paper are not in accord with the results obtained by Wells, Fair, Whisler and Koller. This discrepancy appears to be due to the selective sampling of air by means of the centrifuge. Experiments indicating this explanation are now being published in the *Journal of Bacteriology* by the writers. Further evidence of the varying susceptibility of bacteria at different phases of the life span is also contained in the article being published.

SEASONAL VARIATION IN REACTIONS TO HOT ATMOSPHERES

By F. C. HOUGHTEN,* A. A. ROSENBERG,** AND M. B. FERDERBER, M.D.,***
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THE Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS has been interested in a study of the physiological reactions of persons to hot atmospheres since 1922, and a number of reports on this work have been published in the TRANSACTIONS of the Society. During 1938 the study was carried on under the Technical Advisory Committee on Air Conditioning in Industry, consisting of A. E. Stacey, Jr., *Chairman,* Philip Drinker, Dr. Leonard Greenburg, H. P. Greenwald, A. M. Kinney, J. W. Kreuttner, L. L. Lewis, Dr. W. J. McConnell, Dr. C. P. McCord, P. A. McKittrick, Dr. R. R. Sayers, Charles Sheard, C. Tasker and R. M. Watt, Jr. The work was directed toward a study of the reactions of workers to hot atmospheres in industry in order to develop data of value in applications of air conditioning in industry. A comprehensive report [1] of this work was presented at the 1939 Annual Meeting, giving such physiological reactions as: rise in body temperature, increase in pulse rate, increase in leucocyte blood count, decrease in vital capacity, degree of perspiration, and the feeling of warmth, after the lapse of certain periods of work in various hot atmospheres.

Since all of the data published in the earlier report were collected during the summer, the work during 1939 was directed toward a study of the reactions of workers to similar atmospheric conditions during other seasons of the year. The results of this phase of the study, made during the winter, spring and summer seasons, are contained in this paper.

The test procedure followed closely that of the earlier study. The same subjects or others of approximately the same physical and mental make-up as those participating in the previous work were used, and included seven college students ranging in age from 17 to 22 years. Each subject was given a medical

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[1] ASHVE RESEARCH REPORT No. 1106—Air Conditioning in Industry, by W. L. Fleisher, A. E. Stacey, Jr., F. C. Houghten and M. B. Ferderber, M.D. (ASHVE TRANSACTIONS, Vol. 45, 1939, p. 59.)

examination to see if his body temperature, pulse, blood pressure (both systolic and diastolic), and basal metabolism were normal. The average value for this latter determination had been found in the past to be approximately 40 large calories per square meter per hour (or 288 Btu per hour for the average sized adult male), and no subject was selected for these tests if his basal was found to be far different from this norm.

With few exceptions, four subjects participated in each of these tests. Usually five days of hot tests were given in sequence, then a series of five days of cooler tests. Some alternation of hot and cool tests was made over the period of a week in order to determine the effect of acclimatization due to frequent exposure to a hot or cool condition. Considerable data were collected during the months of February and March, a smaller amount during April, and a still less amount during the first part of August of the past summer.

As in the earlier study, the subjects were seated at rest in a control room for about 45 min in order to permit their various physiological reactions to approach a norm, before entering the psychrometric room where they engaged in their specified activity for three hours. This activity consisted of the light work connected with the operation of the "chance machine," as described in the previous paper. Metabolisms were taken during the three hour test period to determine that the rate of work was the same as during the tests previously reported. This light work, requiring that the subject be attentive and on his feet about three-quarters of the time, resulted in an average metabolic rate of 76 large calories per square meter per hour (or 546 Btu per hour for the average sized adult male). During the entire rest and test periods the subject was required periodically to observe and record his various physiological reactions, such as body temperature, pulse rate, vital capacity, degree of perspiration, and feeling of warmth. The medical observer at less frequent intervals took observations of each subject's leucocyte count and blood pressure. The subject was weighed nude at the beginning of the test in the control room as well as after the test in order to obtain a measure of the perspiration loss.

In some of the later tests, readings of rectal temperature and leucocyte count were made after the subject had completed his three hours of test and had been again in the control room for a one hour period. The purpose of this was to obtain an indication of how quickly the body returned to normal after being relieved from the hot atmospheric condition. The atmospheric condition maintained during the period of tests was varied throughout the range of 74 ET to 92 ET; and the relative humidity was maintained at either 60 or 90 per cent, the greater number of tests being run at the lower humidity condition.

TEST DATA AND RESULTS

Tabulations and plots of the various physical and physiological reactions were made for each subject for each individual test; and from these data, summary curves were made up to indicate the general trends and pertinent conclusions. In Fig. 1 is shown the relation between the rise in body temperature above normal in degrees Fahrenheit and the effective temperature of the air condition; the filled-in data points being for 90 per cent relative humidity and the others for 60 per cent. The lower curve with no data points is taken from the previous report and represents the average for 60 and 90 per cent relative humidities. The points indicated by small circles are data

taken during the months of February and March and are referred to as *Winter* data; the few points represented by triangles are for August 1939 and serve to corroborate to a small extent the data observed during the previous summer; and the small squares are used to represent data taken during April, a transition month between winter and summer.

It is readily seen from the curve for the February and March data that a person has a greater rise in body temperature when exposed to the same hot atmospheric environment during the winter than is realized during the summer. The transition curve for April which falls between the summer and winter curves serves to indicate that the body undergoes some progressive change in acclimatizing itself during changes of seasons of the year. Whereas the summer curve for the rise in body temperature increases from 0 deg to about 1.3 deg through the range of effective temperatures of 78 to 91, the winter curve
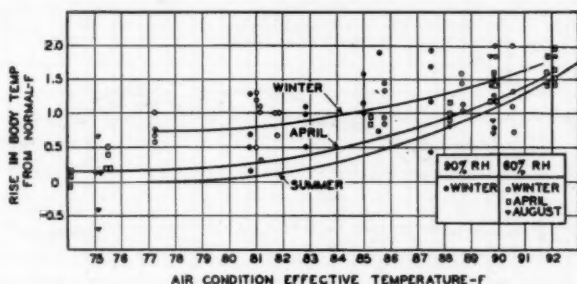
FIG. 1. RELATION BETWEEN RISE IN BODY TEMPERATURE AND EFFECTIVE TEMPERATURE OF INDOOR AIR CONDITION AT 60 AND 90 PER CENT RELATIVE HUMIDITIES, FOR SUMMER, APRIL AND WINTER

increases from 0.75 deg to 1.65 deg; that is, at the lower effective temperature the rise is 0.73 deg higher during the winter, while it is only 0.35 deg or less than half as much higher at the greater effective temperature. This indicates that when the body is exposed to high indoor effective temperatures, acclimatization would have but little effect in regulating the body temperature. The fact that all three curves appear to be converging at the higher effective temperatures indicates that acclimatization has little effect at these extreme conditions of indoor effective temperature.

Another very significant fact is arrived at by closely examining and comparing the summer and winter curves. It can be seen that the 1.0 deg temperature rise ordinate line intersects the winter curve at approximately 84.5 deg ET, whereas it intersects the summer curve at 89.5 deg ET. That is, the abscissae difference between the summer and winter lines at this 1.0 deg body temperature rise is 5 deg ET, or the same difference that occurs between the optimum summer and winter comfort conditions of 71 and 66 deg ET, respectively.

In order to furnish an indication of how well the body responds in returning to normal temperature after being relieved from a hot atmospheric condition to one of fairly ideal condition, one day's temperature data have been plotted

in Fig. 2. Curves as shown for each of four subjects participating in the test during which the condition in the control room was approximately 73 deg ET; and a condition of 88 deg ET and a relative humidity of 60 per cent were maintained in the test room. This figure shows how each subject's temperature was allowed to reach equilibrium during the one hour in the control room prior to entering the test chamber; then during intervals in the test room, temperatures were recorded to indicate how rapidly and to what extent the body temperature increased. The average increase for three hours occupancy on this day was approximately 1.0 deg. After the subjects returned to the control room and remained at rest, all the body temperatures tended to return to normal; the average of the body temperatures was found to be only 0.2 deg above normal at the end of an hour of recuperation, an average drop of 0.8 in an hour. Likewise, in all other days of test the body was found to recuperate
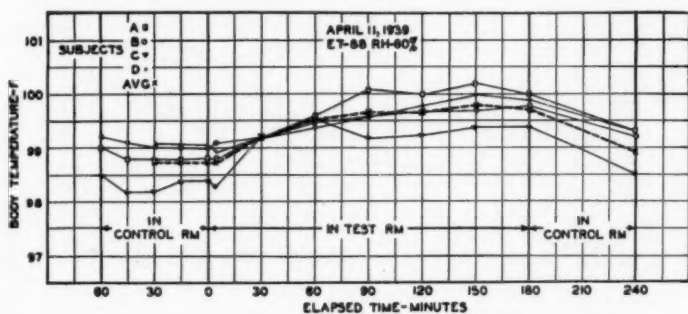
Fig. 2. Relation Between Rise in Body Temperature and Exposure in Control and Psychrometric Rooms

quite well after an hour's rest in the control room, for in no instance did any subject's body temperature fail to return to within 0.4 deg of normal even though the body temperature had risen 2.0 deg above normal after three hours exposure in the hot condition.

The increase in pulse rate above normal in beats per minute for all subjects after three hours exposure in all tests is plotted against effective temperature in Fig. 3, data for both 60 and 90 per cent relative humidity tests being included. These show to very good advantage, in a manner similar to Fig. 1, that acclimatization has a very definite effect on a person's reaction to indoor atmospheric conditions. At the lower effective temperatures the winter curve is displaced an appreciable amount above the summer pulse curve, with the April transition curve falling between the two and the three tending to run together at some high atmospheric condition.

The new leucocyte count data indicate that the summer curve would move slightly upward and to the left for winter months. There is shown in Fig. 4 for each of four subjects, as well as for the average value, the increase in the leucocyte count of the blood in white cells per cubic millimeter during three hours in the hot atmospheric condition, and the later decrease after the subjects' return to the control room. In no day of test did any subject's count
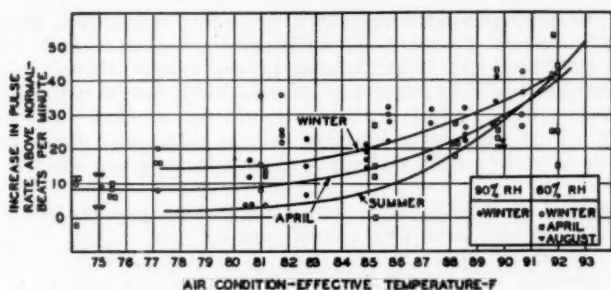
FIG. 3. RELATION BETWEEN INCREASE IN PULSE RATE IN BEATS PER MINUTE AND EFFECTIVE TEMPERATURE OF INDOOR AIR CONDITION AT 60 AND 90 PER CENT RELATIVE HUMIDITIES, FOR SUMMER, APRIL AND WINTER

fail to return toward normal, but it was found that usually one hour was insufficient time to bring the count completely to normalcy; that is, in most instances the count dropped only a little more than halfway back to the initial value.

The ability of the human body to adjust itself to changes in environment is a very important physiologic factor. This flexibility is well illustrated in a comparison of reactions not only to high temperatures but also to invading disease. In fever therapy research, it has been shown that long frequent exposures to hot atmospheres produce some changes in leucocyte response. The first treatments produce a distinct rise in white blood cells, but this rise may be considerably reduced or absent in successive treatments if the interval elapsing between treatments is insufficient.

While this explanation may be theoretical it is proposed as probable that there is a state of exhaustion or fatigue of the hemopoetic system which produces the white cells so necessary for protection against invading infection.

This explanation may be augmented by knowledge of what occurs to a person suffering from a chronic infection. At this point one must consider
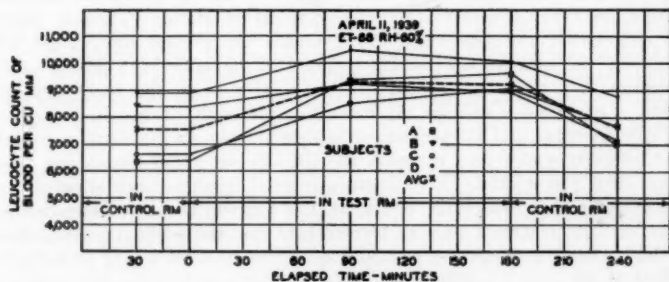
FIG. 4. RELATION BETWEEN INCREASE IN LEUCOCYTE COUNT AND EXPOSURE IN CONTROL AND PSYCHROMETRIC ROOMS

also the temperature mechanism that is involved. If a person develops an infection, his natural defensive response is characterized by a rise in body temperature and leucocytes. If the infection becomes chronic, there is less rise in either until eventually the count may be numerically normal and the temperature shows only the diurnal variation. Should this condition exist for some time and a more fulminating infection, such as pneumonia, invade the body, the failure of those necessary elements renders the body less effective against the illness.

Exposure of workers to high effective temperatures may produce the same changes if continued over long periods without some respite, in the form of cool surroundings as is experienced during the winter months. The rise in body temperature (Fig. 1) demonstrates this difference and, if a greater mass of summer data were available, one might expect to find the same tendencies in the leucocyte change. Another illustration is shown by the seriousness of so called "summer pneumonia" despite the fact that patients may receive the same treatment as during the pneumonia seasons. The stormy course and failure of therapeutic measures, in some measure, may be traced to the exhaustive state of the protective temperature and leucocyte rise due, presumably, to acclimatization.

From workers who have spent long periods in certain very hot climates, there is evidence to show that the failure of physiologic defense against infections is directly proportional to the time spent in the hot area. If true, one might conclude that the constant existence in extremely hot environments might produce a condition in the body where both temperature and white blood count may fail to increase as a defense measure.

This entire subject of temperature rise and increase in leucocyte count, and their significance in health and in disease make it highly important that active laboratory investigation of the subject be pressed forward.

No new data are given for vital capacities, blood pressures, weight losses, since these reactions show no pronounced change over those observed during the summer.

SUMMARY

From the data presented in this paper it is indicated that a person's acclimatization to atmospheric environments during different seasons of the year not only affects his desire for some higher or lower effective temperature condition, but also affects certain physiological reactions. That is, both his body temperature rise and pulse rate increase are greater during the winter than during the summer upon being exposed to the same high indoor effective temperature for the same period of time. Moreover, the body appears to undergo a progressive change in acclimatization with change in seasons; for pulse rate increase and temperature rise data during spring show a transition period between winter and summer, the rises being greater than for summer but less than for winter. The charts presented in this paper show also that acclimatization has but little effect at the higher effective temperature in regulating body reactions such as temperature rise and pulse increase; at some quite high effective temperature condition acclimatization would probably have no effect at all.

The sample curves for recovery of body temperature and leucocyte count after a subject has been relieved from a hot atmospheric condition to one of a

comfortable environment indicate that the body returns quite rapidly toward normal. After one hour of recuperation, the body temperature was found to return at least to within 0.4 deg of normal; while the leucocyte count was found to return more than halfway to its initial control room value.

DISCUSSION

W. L. Fleisher: On the effective temperature vs. the increase in pulse rate, you will notice in the illustration that the curves cross at a particular point, and I would like to ask whether that is just the inconsistency of the drawing or whether that is something significant.

F. C. Houghten: The variations in physiological reactions of different individuals under the same stimulus of the same environment including the atmosphere, as well as the variations in a given individual with time and other variables makes it impossible to plot very precise curves, at least, unless a tremendous mass of test data is available for analysis by statistical methods. Hence, it cannot be accurately said that the small variation in the curves as drawn is real or whether these variations may be due to insufficient data. Based on the curves as drawn from the data available, it appears that for moderately hot conditions there is a difference between the winter and summer reactions while for extremely hot conditions the more violent reactions are more or less independent of season.

C.-E. A. Winslow: I must say a word about this. I think this is a very valuable contribution. The one practical conclusion from it is that a given temperature maintained indoors in winter is considerably worse than the same thing in summer. This phenomenon of acclimatization has been studied somewhat, and in our laboratory we have come to the conclusion that this summer-winter difference is due to the summer education of the sweat mechanism. During the summer people sweat much more readily. As Mr. Houghten says, when it gets us to the ultimate point, that will not help them, but they do respond to a hot condition more readily by sweating at an earlier point.

John James (Written): The results of this paper might possibly offer a clue to the debatable question as to whether indoor design temperatures should be lower for comfort air conditioning applications in northern United States as contrasted with the southern region.

Assuming that people living in the southern belt of the country have their protective temperature in an exhaustive condition, it is possible that their demands for higher summer indoor temperatures can be traced to this cause.

It has been always difficult to account for a difference of 5 deg ET between the summer and winter lines on the ASHVE Comfort Chart. The results reported in this paper to the effect that the body appears to undergo a progressive change in acclimatization in different seasons may account for this difference.

ANALYSIS OF THE FACTORS AFFECTING DUCT FRICTION

By J. B. Schmieler,* F. C. Houghten,** and H. T. Olson,*** Pittsburgh, Pa.

T HE results of a study of frictional resistance to the flow of air in ducts were presented at the 45th Annual Meeting of the Society in January, 1939. This paper [1] contained air flow-friction data applying to straight round ducts with and without joints as found under experimental conditions. Since the publication of these findings the Laboratory has been interested in a mathematical analysis of the friction flow relationships, with a view of developing mathematical formulae which may be used in satisfactorily expanding the test data from a few sizes of ducts to a wide range of duct sizes and flow conditions. This analysis, based upon a modified form of the Fanning equation and the friction factor—Reynolds Number relationships for ducts with no joints and ducts with 40 joints per 100 ft length of run, is presented herein.

The loss of pressure due to the frictional resistance to flow of fluids in pipes and ducts is often expressed by the Fanning formula as:

$$H = \frac{fLV^2}{2gD} \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots (1)$$

where H = pressure loss in feet of the fluid
  f = friction factor
  L = length of duct in feet
  V = velocity in feet per second
  g = acceleration due to gravity = 32.2 fps²
  D = diameter of duct in feet

In this formula the friction factor, f is expressed in terms of the Reynolds Number and for ducts or pipe of given characteristics as regards to material and smoothness of construction this factor has generally been assumed to be constant, regardless of size. An average value of 0.0149 taken from the test

results for the friction factor in the earlier Laboratory report [2] led to the development of the following formula:

$$H = \frac{0.765 fL \ (V/4000)^{1.84}}{D^{1.26}} \quad \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots (2)$$

where the different symbols, except for H which is given in inches of water and for V which is expressed in feet per minute, have the same significance as indicated for the Fanning formula previously stated. For ducts with 40 joints per 100 ft length of run the average value of the friction factor as taken from the test data was 0.0192, giving the following:

$$H = \frac{0.75 fL \ (V/4000)^{1.84}}{D^{1.31}} \quad \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots (3)$$

Equations (2) and (3), resulting from a ready analysis of the Laboratory's data as presented in the earlier report, offer an opportunity for an extended analysis as regards the numerical value of the velocity exponent and to a greater extent as regards the evaluation of the friction factor f.

The velocity exponent 1.84 was taken as the numerical average of the observed values for a large number of tests. These values have since been subjected to a statistical treatment, wherein the results of a number of tests on round, shop-made ducts without joints were plotted on log-log paper, and the velocity exponents in the equations showing variation of pressure drop with velocity were measured and found to be as follows:

4-In. Round	8-In. Round	24-In. Round
1.75	1.92	1.64
1.85	1.90	1.83
1.81	1.88	1.89
1.68	1.81	1.80
. . .	1.87	1.81
. . .	1.91	. . .
. . .	1.86	. . .
. . .	1.87	. . .
. . .	1.79	. . .
. . .	1.80	. . .
. . .	1.80	. . .
. . .	1.87	. . .

These exponents were found to have the average value of 1.826 and when treated statistically the standard deviation σ was found to be 0.0698, indicating that the most probable value for the average velocity exponent is 1.826 ± 0.027. This means that 9 groups out of 10, each containing approximately the same number of observations as this group, will yield an average value of the velocity exponent within the limits given. Although some authorities have used an exponent of 2.0, it can be seen that the test results never gave a value above 1.92. Most recent investigators show a fractional velocity exponent for ducts of roughness comparable to sheet metal. Noteworthy among these

[2] Loc. Cit. Note 1.

is the work of Kemler [3] in which are found data by numerous experimenters throughout the country on resistance to flow of various fluids through many types of pipes and ducts. In nearly all cases, the velocity exponent was lower than 2.0. Kemler's work shows velocity exponents by various investigators for galvanized ducts of 1.67, 1.77, 1.82, 1.83, 1.85, and 1.89. The 1938 edition of *Fan Engineering* gives 1.82 for smooth, straight galvanized ducts and clean steel and wrought iron pipes. THE ASHVE GUIDE for the past number of years has shown a value of 1.86 for the exponent. These disagreements may be due to variations in the test set-up, physical differences in the material used, method of analysis, nature of turbulent flow, and other factors.

Fig. 1 shows the relation between the friction factor and the Reynolds Number for the three sizes of round ducts studied at the Laboratory and presented in the earlier paper.[4] As pointed out therein it is seen that rather
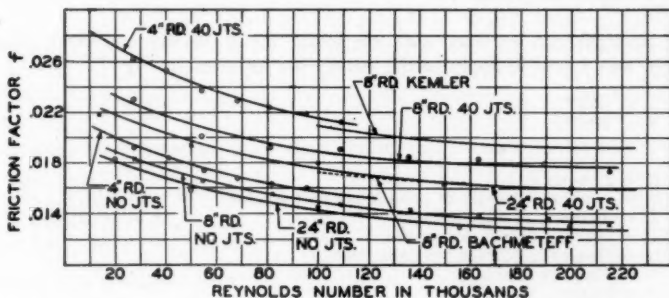
FIG. 1. RELATION BETWEEN THE FRICTION FACTOR AND REYNOLDS NUMBER FOR ROUND DUCTS TESTED WITHOUT JOINTS AND WITH 40 JOINTS PER 100 FT LENGTH OF RUN. CURVES FOR DATA PUBLISHED BY KEMLER [5] AND BACHMETEFF [6] ARE INCLUDED

than being independent of duct size definite curves result for each duct size. The reason for this lies in the fact that the roughness of the sheet metal is more or less constant rather than proportional to the duct size. Also, it will be seen that the shape of the curves makes it impossible to use a single value of f in equation (1) for calculating pressure drops, even for a single size of duct of any particular design, but that this value must be some function of the Reynolds Number. This suggests the possibility of expressing f as some function of the Reynolds Number and duct size, or as some function of the two expressions.

The friction factor—Reynolds Number relationship is commonly expressed as

[3] Study of Data on the Flow of Fluids in Pipes, by E. Kemler. (*ASME Transactions*, Hyd., Vol. 55, 1933.)
[4] Loc. Cit. Note 1.
[5] Loc. Cit. Note 3.
[6] The Reynolds Number, by B. A. Bachmeteff (*Mechanical Engineering*, October, 1936).

$$f = \frac{K}{Re^n} \text{ or } f = C + \frac{K}{Re^n} \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots (4)$$

where

f = friction factor
C = a constant
K = a constant
n = exponent (value to be determined)
Re = Reynolds Number (dimensionless)

where

$$Re = \frac{DV\rho}{\mu} = \frac{DV}{\nu}$$

D = diameter of duct in feet
V = velocity in feet per second
ρ = density of fluid in pounds per cubic foot
μ = absolute viscosity in pounds per foot second

ν = kinematic viscosity = $\frac{\mu}{\rho}$

For straight run of duct with no joints the following empirical friction formula was found to apply to the curves for all sizes in Fig. 1:

$$f = \frac{K\mu^m}{D^x V^y \rho^s} \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots (5)$$

where

K = a constant
μ = absolute viscosity in pounds per foot second
D = diameter of duct in feet
V = velocity in feet per second
ρ = density in pounds per cubic foot
m = viscosity exponent
x = diameter exponent
y = velocity exponent
s = density exponent

By maintaining all factors constant in Equation (5) and varying velocity, density, and viscosity separately and then plotting the values of f against

TABLE 1.   Values of x and K in Equation (6) Based Upon
the Diameters   of Duct and Velocities Indicated
(*No Joints*)

AIR VELOCITY FPM	VALUES BASED ON 8-IN. AND 24-IN. DUCTS		VALUES BASED ON 4-IN. AND 8-IN. DUCTS	
	x	K	x	K
800	0.183	0.1045	0.214	0.1033
1200	0.177	0.1052	0.214	0.1036
1600	0.187	0.1041	0.226	0.1024
2000	0.177	0.1057	0.214	0.1042
2400	0.217	0.1037	0.226	0.1034
2800	0.230	0.1034	0.226	0.1035
3200	0.247	0.1028	0.226	0.1037

Average value x = 0.212.   Average value K = 0.104.

velocity, density, and viscosity, respectively, the exponents, m, y, and z were each found to be equal to approximately 0.17.

The value of x, the diameter exponent, may be obtained through the substitution of two sets of values in the following equation, based upon Equation (5), and the substitution therein of the values found for m, y and z:

$$f = \frac{K\mu^{0.17}}{D^x V^{0.17} \rho^{0.17}} = \frac{K\nu^{0.17}}{D^x V^{0.17}} \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots (6)$$

Substituting the following values for an 8-in. duct where

$V = 2400$ fpm
$\nu =$ (kinematic viscosity at standard conditions)
$\quad = 16.27 \times 10^{-5}$
and $f = 0.01374$ (test data)

Equation (6) becomes

$$0.01374 = \frac{K(16.27 \times 10^{-5})^{0.17}}{0.667^x (2400/60)^{0.17}} \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots (7)$$

Solving for K

$$K = \frac{0.01374 \, (0.667)^x \, (2400/60)^{0.17}}{(16.27 \times 10^{-5})^{0.17}} \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots (8)$$

Substituting the following values for a 24-in. duct where

$D = 2$ ft
$V = 2400$ fpm
$\nu = 16.27 \times 10^{-5}$
and $f = 0.01083$ (test data)

Equation (6) becomes

$$0.01083 = \frac{K(16.27 \times 10^{-5})^{0.17}}{2^x (2400/60)^{0.17}} \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots (9)$$

and substituting K of Equation (8) in (9) the following is obtained:

$$0.01083 = \left(\frac{0.01374 \, (0.667)^x \, (2400/60)^{0.17}}{(16.27 \times 10^{-5})^{0.17}}\right)\left(\frac{(16.27 \times 10^{-5})^{0.17}}{2^x (2400/60)^{0.17}}\right) \dots\dots (10)$$

which reduces to

$$0.01083 = 0.01374 \, (0.333)^x$$

or

$$x = \frac{\log 0.788}{\log 0.333} = \frac{-0.103}{-0.478} = 0.217$$

Using this value of x in solving for K in Equation (8)

$$K = \frac{0.01374 \, (0.667)^{0.217} \, (2400/60)^{0.17}}{(16.27 \times 10^{-5})^{0.17}} = 0.104 \dots\dots\dots\dots\dots\dots\dots (11)$$

Table 1 gives values for x and K from calculations based upon 14 different combinations of 2 sizes of ducts with a given velocity, showing an average

TABLE 2.   VALUES OF x AND K IN EQUATION (6) BASED UPON
THE DIAMETERS OF DUCT AND VELOCITIES INDICATED
(40 Joints per 100 ft.)

AIR VELOCITY FPM	VALUES BASED ON 8-IN. AND 24-IN. DUCTS		VALUES BASED ON 4-IN. AND 8-IN. DUCTS	
	x	K	x	K
800	0.241	0.1332	0.287	0.1306
1200	0.249	0.1332	0.299	0.1305
1600	0.266	0.1334	0.287	0.1322
2000	0.307	0.1330	0.275	0.1345
2400	0.312	0.1332	0.275	0.1350
2800	0.320	0.1323	0.292	0.1341
3200	0.327	0.1313	0.276	0.1341

Average value $x = 0.287$.   Average value $K = 0.133$.

value for x and K of 0.212 and 0.104, respectively.   These values (rather than those determined previously from a single combination of diameters and velocity), will be used.

The complete friction factor equation may then be expressed as

$$f = \frac{0.104 \; \mu^{0.17}}{D^{0.212} \; V^{0.17} \; \rho^{0.17}} \quad \dots \dots \dots \dots \dots \dots \dots \dots \dots \dots \dots \dots \dots \dots (12)$$

To check the accuracy of this equation the calculated results are compared with test results in Table 3.   From this it can be seen that the calculated results check the test data fairly well for the three sizes of ducts at the given velocities. The maximum variation occurs for the 24-in. round duct at a velocity of 800 fpm, where the variation is 2.7 per cent.   This variation is not large considering the chances for error in measurements at the low velocity.   At higher velocities the calculated results check the test data very closely.

Hence, it would appear that a single equation would satisfactorily express the friction factor relationship for the three sizes of ducts listed, and for intermediate sizes, with an accuracy of at least 3 per cent.

A friction loss formula, which excludes the Reynolds Number and the friction factor entirely, may be developed by substituting the value of the friction factor, Equation (12) in the Fanning Equation (1).   Changing H to inches of water and V to feet per minute, Equation (1) becomes

$$H = \frac{fL(V/4000)^2 S}{D} \quad \dots \dots \dots \dots \dots \dots \dots \dots \dots \dots \dots \dots \dots \dots (13)$$

where

$$S = \frac{\text{density of air at test conditions}}{\text{density of air at standard conditions}}$$

Substituting Equation (12) for f in Equation (13) the friction loss formula becomes

$$H = \frac{0.104 \; \mu^{0.17} L(V/4000)^2 \; S}{D^{0.212} \; V^{0.17} \; \rho^{0.17} D}$$

$$= \frac{0.104 \ \mu^{0.17} \ L \ V^2 \ (1/4000)^{1.83} \ (1/4000)^{0.17} S}{D^{0.212} V^{0.17} \rho^{0.17} \ D} \quad \dots\dots\dots\dots\dots(14)$$

Changing $V^{0.17}$ from feet per second to feet per minute, and substituting $S \cdot \rho$ std. for ρ and collecting terms, the equation becomes:

$$H = \frac{2.01 \ (0.104)(0.244) \ \mu^{0.17} \ L(V/4000)^{1.83} S^{0.83}}{D^{1.212} \rho \ \text{std.}^{0.17}} \quad \dots\dots\dots\dots(15)$$

For air at standard conditions and $L = 100$ ft, the equation reduces to:

$$H = \frac{1.157(V/4000)^{1.83}}{D^{1.212}} \quad \dots\dots\dots\dots\dots\dots\dots\dots(16)$$

It should be emphasized that the above equations apply only to straight, round sheet metal ducts with no joints.

In Table 4 there are included a number of calculated values as well as test data values for the 4-in., 8-in. and 24-in. duct sizes which show the degree of application. Column B for each of the three sizes of duct gives the calculated pressure loss using the new formula, for the velocity given in Column A. Column C shows the pressure loss determined by the new formula in per cent of the average test data. It will be noted that for the three sizes of duct the average of the calculated results is 99.7 and 100.6 per cent for the first two cases and 99.6 per cent for the 24-in. duct size, with maximum variations of

TABLE 3.   RELATION BETWEEN CALCULATED FRICTION FACTOR AND TEST FRICTION FACTOR FOR STRAIGHT RUN OF DUCT WITH AND WITHOUT JOINTS

VELOCITY FPM	FRICTION FACTOR				PER CENT DIFFERENCE	
	Test		Calculated			
	A	B	A	B	A	B
4–IN. ROUND						
800	0.01915	0.0262	0.01923	0.02678	0.42	2.21
1600	0.01702	0.0235	0.01705	0.02375	0.18	1.05
2800	0.01564	0.0218	0.0155	0.02159	−0.90	−0.96
8–IN. ROUND						
800	0.0165	0.0215	0.01656	0.02192	0.36	2.00
1600	0.0146	0.0193	0.01468	0.01943	0.55	0.67
2800	0.0134	0.0178	0.01336	0.01766	−0.30	−0.79
24-IN. ROUND						
800	0.0135	0.0165	0.01314	0.01596	−2.67	−3.38
1600	0.01188	0.0144	0.01166	0.01416	−1.86	−1.67
2800	0.01041	0.0125	0.01059	0.01287	1.72	3.00

(A) For duct with no joints.   (B) For duct with 40 joints per 100 ft.

TABLE 4.   RELATION BETWEEN PRESSURE DROP IN INCHES OF WATER PER 100 FT AND VELOCITY FOR STRAIGHT RUN OF DUCTS—NO JOINTS

				SHOP MADE DUCT					FACTORY MADE		
A	B	C	D	E	F	G	H	I	J	K	L
Velocity Fpm	Calculated New Lab. Eq.	Column B in Per Cent of E	Calculated—Old Lab. Eq. in Per Cent of E	Avg Test Data	High Value Curve	Column F in Per Cent of E	Low Value Curve	Column H in Per Cent of E	Low Value Curve in Per Cent of E	Avg Test Data in Per Cent of E	High Value Curve in Per Cent of E
4-IN. ROUND				**INCLUDES 4 TESTS**					**INCLUDES 4 TESTS**		
400	0.065	100.0	101.7	0.065	0.070	107.6	0.060	92.4		132.2	
800	0.230	100.0	103.7	0.230	0.240	104.3	0.218	94.7		113.8	
1200	0.487	100.4	102.7	0.485	0.485	100.0	0.485	100.0	Same As Avg	111.0	Same As Avg
1600	0.816	99.5	103.6	0.820	0.840	102.4	0.820	100.0		111.2	
2000	1.23	99.2	103.1	1.24	1.265	102.1	1.24	100.0		108.8	
2400	1.73	100.0	103.3	1.73	1.768	102.1	1.73	100.0		107.2	
2800	2.28	99.1	103.0	2.30	2.36	102.5	2.30	100.0		104.3	
3200	2.92	99.3	102.6	2.94	3.02	102.5	2.94	100.0		100.3	
Avg	...	99.7	102.9	...	...	102.9	...	98.4		111.1	
8-IN. ROUND				**INCLUDES 12 TESTS**					**INCLUDES 4 TESTS**		
400	0.028	103.7	99.7	0.027	0.027	100.0	0.027	100.0	96.4	100.0	103.3
800	0.100	101.0	100.5	0.099	0.114	115.2	0.085	95.8	98.0	100.0	102.4
1200	0.210	101.0	99.3	0.208	0.232	111.5	0.168	80.8	100.0	101.3	102.0
1600	0.351	100.0	100.2	0.351	0.391	111.3	0.281	80.0	101.2	102.4	103.5
2000	0.531	99.6	100.3	0.533	0.590	110.8	0.442	83.1	100.4	101.3	102.2
2400	0.744	100.1	99.6	0.743	0.830	111.7	0.645	86.8	102.3	105.0	107.5
2800	0.982	99.7	100.5	0.985	1.11	112.8	0.873	88.6	100.4	103.6	106.6
3200	1.26	100.0	100.1	1.260	1.42	112.5	1.435	90.2	102.3	104.6	106.8
Avg	...	100.6	100.0	...	...	110.7	...	88.2	100.1	102.3	104.4
24-IN. ROUND				**INCLUDES 5 TESTS**							
800	0.026	96.3	93.0	0.027	0.0275	101.7	0.029	107.5			
1200	0.055	96.5	91.5	0.057	0.057	100.0	0.058	101.6			
1600	0.093	97.9	93.0	0.095	0.097	102.0	0.094	98.9			
2000	0.140	100.0	95.1	0.140	0.143	102.0	0.134	95.8	No Tests Taken		
2400	0.197	101.0	95.4	0.195	0.20	102.5	0.182	93.8			
2800	0.259	101.6	97.3	0.255	0.270	105.8	0.237	93.0			
3200	0.332	103.8	98.7	0.320	0.335	104.7	0.295	92.2			
Avg	...	99.6	94.9	...	...	102.7	...	97.5			

Notes on column headings:

(B) Calculated, based on new Lab. equation, $H = 1.157 \ (V/4000)^{1.89}/D^{1.212}$

(C) Calculated, based on old Lab. equation, $H = 1.14 \ (V/4000)^{1.84}/D^{1.26}$

0.9, 3.7 and 3.8 per cent, respectively.   This is certainly a very satisfactory agreement between theoretically and experimentally determined values when the range of velocities and duct sizes are considered.   Column D shows the pressure loss calculated by the old Laboratory formula in per cent of average test data. It will be noted that the average of the calculated values, 102.9, 100, and 94.9 per cent for the three sizes, is given.   Columns G and I show variation in pressure loss for curves giving high and low values in per cent of the average test data.   It should be emphasized that these variations represent extreme conditions and that few tests gave this high variation.   It should also be noted that these variations may result from a wide range of factors, including errors in making physical observations, the accuracy of the instruments used, leaks in long rubber tubing connections from the static pressure taps to the manometers, air leakage from the ducts in which measurements were made, and the many possible variations in the construction of the duct itself and its erection.   In the 8-in. duct study undoubtedly much of the variation is due to the many different duct set-ups used; that is, different tests in a series included duct which had been assembled and dismantled a number of times, including all the attendant misuse.   In all such assembling and dismantling due care was taken to avoid misuse of the duct, but nevertheless some misuse must be assumed to have occurred.   Considering all these factors, it is obvious that the agreement between the experimental and theoretical data is much better than could be hoped for in the case of practical duct installations as designed and erected throughout the country under the same set of specifications.   In other words, individual workmanship, choice of materials, and erecting characteristics will probably account for greater variations than are included in the results of this study.   In the last three columns, J, K, and L, for the 4-in. and 8-in. round ducts, are shown some comparisons between factory-made duct and the average shop-made duct values.   This shows a maximum variation in pressure loss of 4.6 per cent for the 8-in. round duct and 13.8 per cent (excluding 400 fpm velocity) for the 4-in. round duct.

The friction factor equation for 40 joints per 100 ft length of run was found by similar calculations.   Different values for x and K in Equation (6) based upon different combinations of duct diameters and velocity are given in Table 2, indicating average values for x and K of 0.287 and 0.133, respectively.   This gives the following equation for the friction factor:

$$f = \frac{0.133\ \mu^{0.17}}{D^{0.287}V^{0.17}\rho^{0.17}} \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots(17)$$

Table 3 shows the rather close agreement between test data and calculated results based on this formula.   The maximum variations are 2.2, 2.0, and 3.4 per cent for the three sizes of duct at the velocities given.

Substituting the value of f from Equation (17) in Equation (1) the friction loss formula for air at standard conditions and for $L = 100$ ft reduces to:

$$H = \frac{1.48\ (V/4000)^{1.83}}{D^{1.287}} \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots(18)$$

A comparison of calculated results with test results is presented in Table 5 to check the application of this formula in calculating friction losses.   The calculated results of other investigators are also shown for comparison.

TABLE 5. RELATION BETWEEN PRESSURE DROP IN INCHES OF WATER PER 100 FT AND VELOCITY FOR STRAIGHT RUN OF DUCT—WITH JOINTS

A	B	C	D	E	F	G
Velocity FPM	Calculated New Lab. Eq.	Column B In Per Cent of G	Calculated Old Lab. Eq. In Per Cent of G	Calculated Zwickl Eq. In Per Cent of G	Calculated Fan Engr. Eq. In Per Cent of G	Avg Test Data

4-IN. ROUND

A	B	C	D	E	F	G
400	0.091	102.2	99.8	141	108	0.089
800	0.320	101.6	102.3	146	109	0.315
1200	0.676	100.9	99.3	145	107	0.670
1600	1.13	100.0	100.0	147	107	1.13
2000	1.71	100.6	101.2	147	108	1.70
2400	2.40	100.0	99.6	146	107	2.40
2800	3.17	99.1	99.7	146	107	3.20
3200	4.05	100.2	100.6	149	108	4.04
Avg	...	100.6	100.3	146	108	...

8-IN. ROUND

A	B	C	D	E	F	G
400	0.037	104.2	100.0	145	113	0.0355
800	0.131	101.6	100.0	146	113	0.129
1200	0.277	102.2	99.0	147	111	0.271
1600	0.465	100.4	98.4	147	111	0.463
2000	0.701	99.7	97.8	146	110	0.703
2400	0.983	99.8	97.5	146	110	0.985
2800	1.31	100.0	99.3	147	110	1.31
3200	1.66	100.0	98.8	148	111	1.66
Avg	...	101.0	98.9	146	111	...

24-IN. ROUND

A	B	C	D	E	F	G
800	0.032	97.0	93.9	139	113	0.033
1200	0.067	97.1	92.3	140	112	0.069
1600	0.112	97.4	92.9	144	114	0.115
2000	0.170	101.2	97.0	148	119	0.168
2400	0.238	101.7	97.0	150	119	0.234
2800	0.314	102.3	98.7	152	121	0.307
3200	0.401	103.9	100.0	155	123	0.386
Avg	...	100.1	96.0	147	117	...

Notes on column headings:

(B) Calculated, based on new Laboratory equation,
$H = 1.48 \ (V/4000)^{1.83}/D^{1.387}$ with 40 joints/100 ft.

(D) Calculated, based on old Laboratory equation,
$H = 1.44 \ (V/4000)^{1.84}/D^{1.31}$ with 40 joints/100 ft.

(E) Calculated, based on Zwickl equation,
$H = 2.2 \ (V/4000)^{1.87}/D^{2.0}$ with unstated joint freq.

(F) Calculated, based on new Fan Engineering equation,
$H = 1.678 \ (V/4000)^{1.84}/D^{1.36}$ with unstated joint freq.

Column *B* shows the calculated pressure drop per 100 ft using the new formula for the velocities in Column *A*. Column *C* gives this pressure drop in per cent of the average test data values in Column *G*. It will be noted that for the three sizes of duct average percentages of 100.6, 101.0 and 100.1 per cent are given. The maximum variation from average test data is 2.2 per cent for the 4-in., 4.2 per cent for the 8-in. and 3.9 per cent for the 24-in. duct. Thus, it is conclusively shown that the calculated results based on the formula presented here check the average values from test data, and the amount of variation shown is negligible when all factors are taken into consideration. Column *D* gives the calculated pressure drop based on the old formula in per cent of the average test data values. It will be noted that the average of these figures is 100.3 and 98.9 per cent for the first two sizes and 96 per cent for the 24-in. duct, and the calculated results based on other sources are also given for comparison. In Column *E* are pressure losses based on the formula by Dr. Zwickl [7] in per cent of average test data, while Column *F* gives the calculated pressure drop based on the formula given in the 1938 edition of *Fan Engineering* in per cent of average test data values. It should be stated that the data of Zwickl and *Fan Engineering* are for duct of an unstated joint frequency while the Laboratory data are based on duct with 40 joints per 100 ft.

It is well to point out that the Fanning Equation (1), using the square power of the velocity, will give identical results as obtained by using the derived friction loss formulae with fractional exponents. However, at the velocity for which the pressure loss is desired, the corresponding friction factor would either have to be chosen from a curve or calculated, and this value substituted in the Fanning equation. This would necessitate extra curves and computation. The Laboratory formulae here presented, although embodying fractional exponents, eliminate friction factor and Reynolds Number entirely and offer a more complete expression for frictional resistance.

Conclusions

1. Analysis of the test data indicates a value of 1.83 as the exponent of the velocity which fits test data best. The velocity exponent of 2 which has been generally accepted and used by many engineers does not fall within the entire range of the test data.

2. Calculated pressure losses both for duct with no joints and duct with 40 joints per 100 ft, using the friction loss formulae developed in this paper, check test data to a high degree of accuracy. The close agreement between theoretical and experimental results shown for the wide range of velocities and duct sizes studied serves to lend confidence to the acceptance of these formulae in calculating pressure losses for smaller, intermediate, and larger sizes of duct.

DISCUSSION

W. L. FLEISHER: I know from my own experience that there is some opposition to the use of factors which are in this paper, as opposed to those that are in general use. I believe that the meeting is the place for open discussion of the findings of the Research Laboratory on this particular thing, with the idea that if there is no

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[7] The Flow of Air and Its Distribution Through Ducts, by Dr. J. R. Zwickl. (*Heating and Ventilating*, February-July, 1939.)

discussion, that is, if there is no belief in the paper, we should know about it then because it is the idea of the Research Laboratory to carry this work further.

F. C. Houghten: This is one of those papers dealing with a mathematical analysis of the results of the Laboratory study which cannot easily be interpreted and discussed at a meeting of this kind, at least, not in detail. Nevertheless, it is probably one of those papers which will be far more valuable in the Society's Transactions to be used in the future in developing duct friction charts and tables.

J. H. Van Alsburg: This work was suggested and instituted, four years ago. It was brought about because some questions were raised regarding the data appearing in The Guide on friction. Many people in the industry questioned the friction chart as to its accuracy. We attempted to investigate the source of the material as far as its printing is concerned and the correlated data that allowed its printing.

For some unknown reason we could not find the source of all this material nor the individual responsible for correlating the data that appear in the current Guide on friction. We listened to the objections from the Society at large as to the friction chart and started some very basic, difficult and very thorough work. We have a very active committee, which proposed a method and an outline of procedure for carrying on this work. We had to wade through a bibliography that appeared in the previous paper, and as you know, it is two pages of close print.

I would also like to add to that although the first paper was accepted we have had some objections or questions concerning some of the data and the factors therein. The paper given today means that we have rechecked carefully all these data, answered any questions and objections that have arisen, to the satisfaction of the committee, and we hope to your satisfaction.

We have proved twice or more, every point mentioned in this paper that differs with any previous data. When this first paper was given last January at Pittsburgh, you may remember a preamble was given to the paper cautioning the use of these data. This work was so basic, we did not want the Society members to thoughtlessly use the published half of it without the other half of the work being completed.

We still offer this word of caution, not that the data are not complete on round straight pipe, but they are not complete on fittings and some of the rectangular pipes, and we are continuing the rectangular work at the present moment. I mean rectangular ducts of one to four ratios and one to eight ratios.

I think you will find over a period of years that this is one of the most basic, one of the most fundamental papers appearing in our work of many years. What is more, it is going to have the widest application of any paper that we have heard for a long while.

THE PERFORMANCE OF STACK HEADS

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This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the University of Wisconsin.

INTRODUCTION

T HE purpose of this investigation was to study the effect of approach on the discharge of air from duct openings similar to those used in rooms. Elaborate tables are supplied by various grille manufacturers to show the distribution of air from the grille face, but these tables assume good approach to the grille such as occurs when the grille is placed at the end of a straight duct. In actual installations, however, poor approach conditions are frequently encountered as the grille is often immediately preceded by an elbow turn or stack head. In this investigation chief attention was given to the determination of the velocities and directions of the components of the air stream as they issued from the face of the stack head being tested.

DESCRIPTION OF APPARATUS

The arrangement used for testing the various approaches, as shown in Fig. 1, consists of a fan for supplying air, a long duct incorporating air measuring devices, a plenum chamber, a stack or vertical duct upon which various approach turns were mounted, and a table with the necessary gages for measuring pressures. Lack of space does not permit complete description of the apparatus except to indicate that the present tests were conducted on a stack head having a dimension of 14 in. x 6 in. Air quantities were measured with a 12-in. x 5-in. calibrated nozzle of the C-2 form reported in Bureau of Standards Research Paper No. 49, installed in a 12-in. duct and checked with the Pitot tube in the 8-in. section of the supply duct.

AIR METERS

For determining the air velocities at the stack head faces several instruments were utilized, including the kata-thermometer, the propeller type anemometer,

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the heated bulb thermometer and the deflecting vane anemometer. A 12-in. wind tunnel with rounded entrance and a 10 ft radius whirling arm were used for calibrations. The deflecting vane anemometer used directly and with a spot reading tip was most satisfactory and was standardized on in these tests. A further check on the accuracy of the instrument and its use was obtained by comparison with the volumes measured at the nozzle. This same instrument
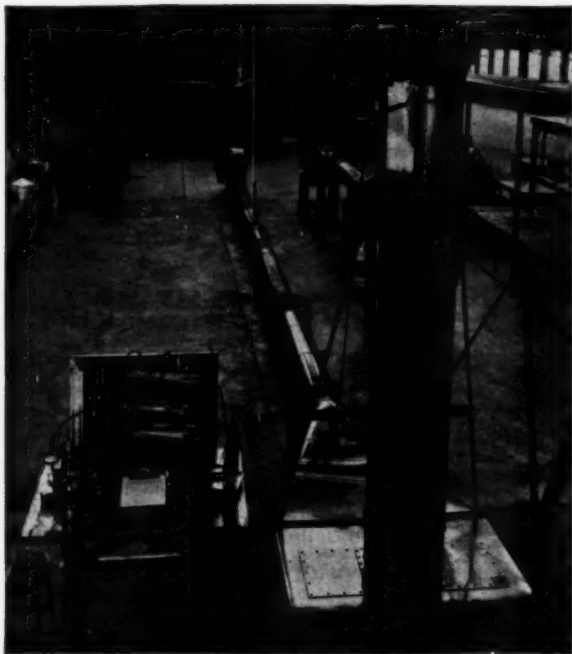
FIG. 1. VIEW OF TEST APPARATUS

with the spot reading tip was also useful along with smoke for determining reverse flow areas.

It was necessary to devise an original instrument to obtain the direction of the air as it leaves the stack head openings. The instrument had to be sensitive enough to indicate the direction of air at the lowest velocities and yet sturdy enough to withstand and record the direction of the highest. The most satisfactory one developed consisted of a carefully balanced balsa wood vane about ¾ in. x 1¼ in., counterbalanced and mounted on a needle shaft with small beads as bearings and supported on a wire yolk and handle. A semi-circle marked in degrees on cardboard was placed at the edge of the stack and a focusing flashlight was used to cast the shadow of the directional indicator on

it. This arrangement shown in Fig. 2 made the angle at which the air left the opening at any point readily determinable.

PROCEDURE

In a test of each stack head observations were made of the direction and velocity at the center of each of nine equal areas of the face, the direction at the four edges of the air stream 12 in. out from the face, the distance up
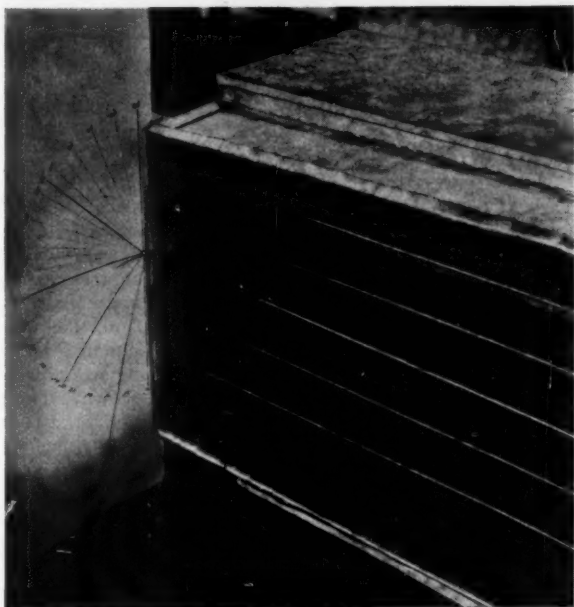
FIG. 2. VIEW OF DIRECTIONAL INDICATOR

from the bottom edge of the face to the point at which zero or reversed flow began; also of the static pressure at the base of the stack head and the pressure readings at the flow nozzle for volumetric determinations. Calculations were made previous to each test to determine the proper gage setting corresponding to 200, 500, 800, 1100 and 1400 fpm stack velocities.

SUMMARY OF RESULTS

In Fig. 3 are shown some of the stack heads tested and Fig. 4 is a drawing of these stack heads and the stack used. The lettered dimensions in Fig. 4 refer to dimensions of the 21 stack heads tested as listed in Table 1. In general the heads classify into two types, Nos. 1 to 10 that have rounded

FIG. 3. STACK HEADS TESTED

backs and 11 to 21 that have square backs. Nos. 1 to 10 classify as round elbow turns and 11 to 21 as square elbow turns.

A summary of test results is given in Table 2 for stack heads 1 to 10 and Table 3 for stack heads 11 to 21. A sketch of the stack head is shown in Column 1, the stack head number, barometer and air temperature are given in Column 2. The relative humidity was not recorded on these tables but was low due to the tests having been made in a space under heated conditions without humidification. The velocity in the 14 in. x 6 in. stack as calculated from the air volume at the flow nozzle is given in Column 3 and the corresponding average face velocity at the stack head opening in Column 4. Stack velocities of 200, 500, 800, 1100 and 1400 fpm are shown. Three sizes of openings were used: 14 in. x 6 in., 14 in. x 9 in., and 14 in. x 12 in. and the
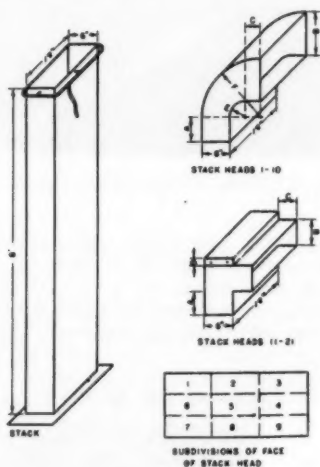
FIG. 4. DIAGRAM OF STACK AND
STACK HEADS TESTED

average face velocity was found by multiplying the stack velocity by the ratio of opening size to stack size.

Columns 5 to 13 inclusive give the tilt upward of the air stream at the center of the nine equal areas of the stack head openings as shown on Fig. 4. A negative reading indicates a tilt downward from the horizontal. In certain cases the angle was indeterminate due to wide fluctuations and these are marked turbulent. Column 14 gives the divergence of the sides of the air stream measured from a vertical plane midway in areas 1 and 3.

The divergence of the air stream boundary 12 in. out from the opening is given in Columns 15 to 18 inclusive, at the top in Column 15, at the bottom in Column 16, at the right side in Column 17 and at the left side in

TABLE 1.   DIMENSIONS OF STACK HEADS TESTED

STACK HEAD No.	A	B	C	D	E	F	T. V.	SPL'T
1	0	6	0	...	3	9	...	0
2	0	9	3	...	3	12	...	0
3	0	12	6	...	3	15	...	0
4	3	6	3	...	0	6	...	0
5	3	9	3	...	0	9	...	0
6	3	12	6	...	0	12	...	0
7	0	6	0	...	6	12	...	0
8	0	6	0	...	3	9	...	2
9	0	9	3	...	3	12	...	2
10	0	6	0	...	6	12	...	2
11	3	6	3	0	...	...	0	...
12	3	6	3	5/8	...	...	0	...
13	3	6	3	1½	...	...	0	...
14	3	6	3	3	...	...	0	...
15	3	6	3	6	...	...	0	...
16	3	9	3	0	...	...	0	...
17	3	9	3	6	...	...	0	...
18	3	12	3	0	...	...	0	...
19	...	6	...	0	3	...	0	...
20	3	6	3	0	...	...	5	...
21	3	9	3	0	...	...	6	...

Column 18. The static pressure at the base of the stack head is shown in Column 19 in inches of water. The location of the four pressure taps is shown in Fig. 4. Static tube traverses at this cross-section indicated agreement with the average from the four static pressure holes. The velocity at the center of each of the nine areas at the plane of the stack head openings are shown in Columns 20 to 28. Negative readings indicate reverse flow into the stack opening. Column 29 gives the arithmetical average of the velocity determined at the center of the nine areas. This may be compared with the average face velocity in Column 4 determined by nozzle measurement. Column 30 indicates the distance up from the bottom edge that negative flow exists. Column 31 shows the static pressure at the base of the stack heads in per cent of the velocity head at the same location.

The results as to directions and velocities of the various parts of the air stream shown in Tables 2 and 3 are shown graphically in Figs. 5 and 6. In

Table 2. Results of Tests on Stack Heads Nos. 1 to 10

these diagrams the velocities found at the face opening were plotted to a convenient scale at an angle equal to the mean angle of the air stream and a smooth curve was drawn through them. Although tests were made at five stack velocities as recorded in Tables 2 and 3, diagrams are shown only for two, namely, 500 and 1100 fpm since they were found to show the behavior adequately. The static pressures at the base of the stack heads are plotted on Figs. 7 to 12 inclusive. In discussing the performance of the stack heads, velocities will be first considered, then directional effects, and lastly static pressure drops.

Stack heads Nos. 1, 2 and 3 were similar having a 3-in. rounded throat radius but having face openings 14 in. x 6 in., 14 in. x 9 in. and 14 in. x 12 in. respectively. A somewhat higher face velocity was found in the middle third of the opening with the 14-in. x 6-in. opening. For the expanding stack heads Nos. 2 and 3 the velocity in the top third was somewhat higher than in the middle third. With the 14-in. x 6-in. opening, a small area of reversed flow, less than an inch in height existed which decreased with an increase in air flow. With the expanding stack heads this area of reverse flow became larger, being 5½-in. in height at the lowest air flow and 4¼-in. at the highest air flow with the 14-in. x 12-in. opening. In all three cases the area decreased with an increase in air flow.

Stack heads Nos. 4, 5 and 6 were similar to Nos. 1, 2 and 3 except that the throats were made square instead of with a 3-in. radius. The performance characteristics of Nos. 4, 5 and 6 were found to be quite similar to those for Nos. 1, 2 and 3 of equal face areas. The tendency for the highest velocities to be in the upper third was more pronounced and the reverse flow areas were larger. In No. 6 with the 14-in. x 12-in. opening, the reverse flow area occupied the entire lower half. With an increase in air flow the reverse flow area increased in No. 4 with a 14-in. x 6-in. opening and remained constant with the expanding stack heads. From the results on these six stack heads it appears that the throat radius is the important factor in determining reverse flow.

Stack head No. 7 which had a 6-in. throat radius gave more uniform velocity distribution over the opening than those with a smaller radius. The area of reversed flow was very small at low velocities and disappeared entirely at the higher velocities.

Stack heads Nos. 8, 9 and 10 were equipped with two splitters each and are otherwise similar to Nos. 1, 2 and 7 respectively. In fact No. 1 became No. 8, No. 2 became No. 9, and No. 7 became No. 10 by the addition of the splitters. The reversed flow area was almost entirely eliminated and the distribution over the face opening was made much more uniform. The effect of the splitters was to make the stack head into three stack heads each with a somewhat higher velocity at the top than at the bottom. This gives a scalloped effect as noted in these three diagrams. In No. 9 which expanded to a 14-in. x 9-in. opening the variation from top to bottom of the individual air streams was more pronounced than in the non-expanding heads Nos. 8 and 10. The short radius turn No. 8 showed performance approximately equal to that of the long radius turn No. 10.

Table 3. Results of Tests on Stack Heads Nos. 11 to 21

Stack heads Nos. 11 to 15 inclusive were tested to determine the influence of varying heights of cushion chambers placed above square type turns. The heights of chambers were 0 in., ⅝ in., 1½ in., 3 in. and 6 in. in order from Nos. 11 to 15. The face opening was 14 in. x 6 in. in all five of these stack heads. The same stack head was used in each case, the change from one to the next in the series being made by substitution of chambers. The purpose of the chamber was considered to be to aid the air in making the turn and thereby make a change from square back characteristics to those of a rounded back. Although the results of these five heads are shown in Table 3, diagrams of only the two extremes, No. 11 with no chamber and No. 15 with a 6-in. chamber are shown in Figs. 5 and 6, since the performance of all was very similar. The velocity was somewhat higher in the top third than in the middle third and this difference increased at the higher velocities and increased a little as the velocity increased. The same comparison was made for the performance of a cushion chamber for stack heads expanding to a 14-in. x 9-in. opening in No. 16 having no chamber and No. 17 with a 6-in. chamber. In each case negative air flow was found in the whole lower third with this area becoming somewhat larger at higher velocities. The velocity distribution was fairly uniform in the upper two thirds with slightly higher velocities in the upper than in the middle third which difference increased with greater air volumes. From a study of the performance of these two series it is concluded that the cushion chamber does not influence distribution.

Stack head No. 18 with a 14-in. x 12-in. opening completes a series of three square turns with No. 11 having a 14-in. x 6-in. opening and No. 16 having a 14-in. x 9-in. opening. The distribution characteristics were quite similar in all three with the highest velocity in the top third, somewhat lower in the middle third, and reverse flow in the lower third. The peak velocities were almost in inverse proportion to the opening area. The reverse flow areas increased somewhat, being at 1100 fpm stack velocity, 32 per cent of the face opening in No. 11, 36 per cent in No. 16 and 39 per cent in No. 18.

Two series of stack heads Nos. 11, 16 and 18, and Nos. 4, 5 and 6 compare square throated turns having rounded backs with those having square backs. No great difference was found, the velocity distribution and reverse flow areas being quite similar in the corresponding stack heads. The largest difference was in the reversed flow area being greater with the rounded back turn No. 6 than with the square back turn No. 18. This difference is accountable by No. 6 having a distance from face to stack (dimension C Table 1) of 6 in. and No. 18 having this distance as 3 in.

Stack head No. 19 was of the square back type similar to No. 11 except that a throat with a 3-in. radius replaced a square throat. The peak velocities were considerably reduced and the area of reversed flow was decreased from nearly two inches to below one inch. The influence of the throat radius is quite apparent.

Stack heads Nos. 20 and 21 were similar to Nos. 11 and 16 except for the addition of turning vanes. In the stack with the 14-in. x 6-in. opening five turning vanes were used and in the stack with 14-in. x 9-in. opening six turning
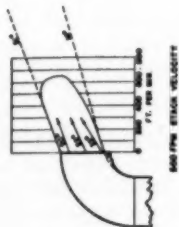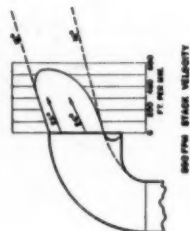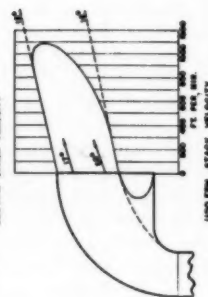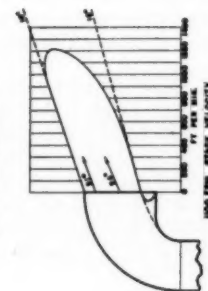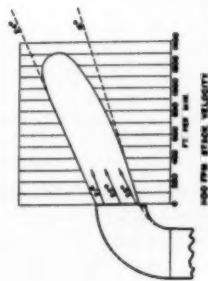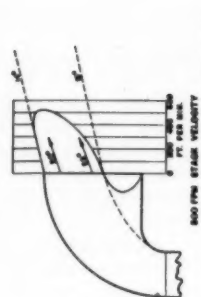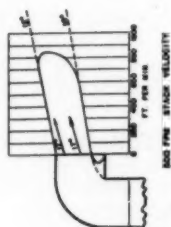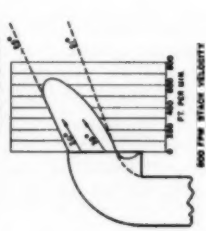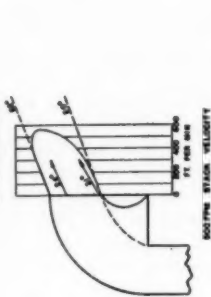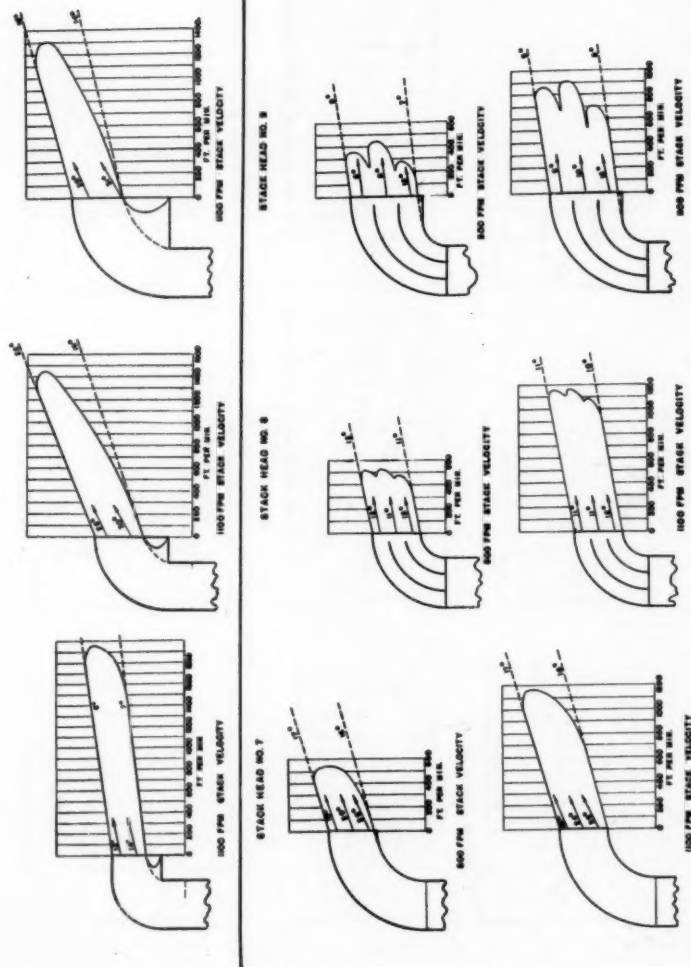
FIG. 5. VELOCITY AND DIRECTION DIAGRAMS FOR STACK HEADS NOS. 1 TO 9
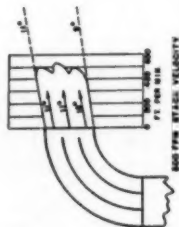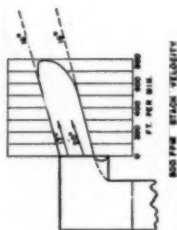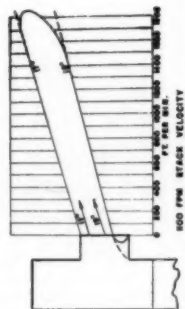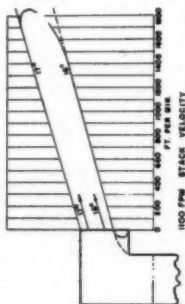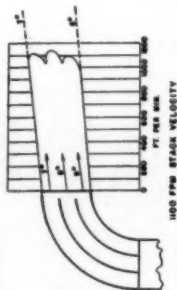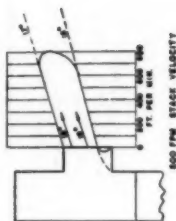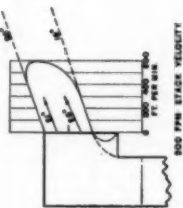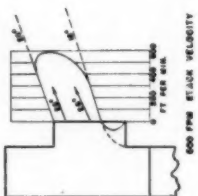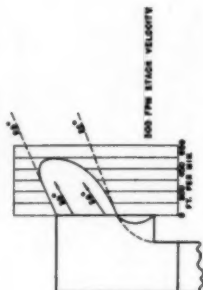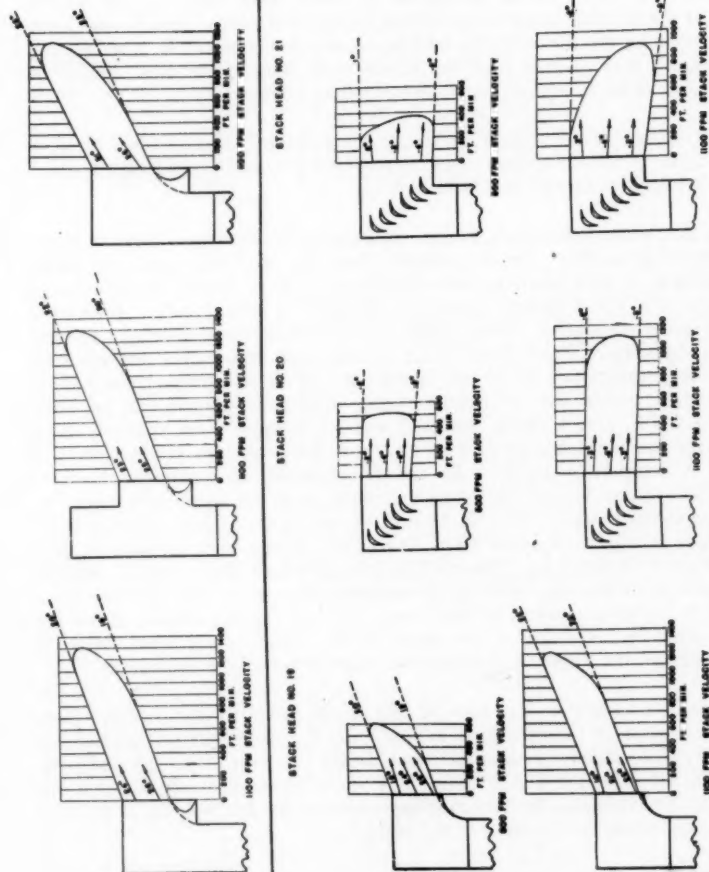
FIG. 6. VELOCITY AND DIRECTION DIAGRAMS FOR STACK HEADS NOS. 10 TO 21

vanes were used all equally spaced. The inside radius of the vanes was 2 in. and the outside radius was 1 in. The turning vanes completely eliminated the reverse flow which extended for almost 2 in. of height in No. 11 and over 3 in. in No. 16. Along with this elimination also went the tendency for higher velocities at the top third of the opening. In fact, the particular installations made resulted in slightly higher velocities at the bottom than at the top. This is probably due to the trailing edges of the vane having a slight dip with the horizontal. A radius on the throat of the stack head equal to that of the outside of the turning vanes and a radius on the back corner of the stack head equal to that on the inside of the turning vanes should result in more perfect distribution. It is evident from the results with stack heads Nos. 20 and 21 that turning vanes in square turns give excellent performance.

The directions of the issuing air stream are tabulated in Columns 5 to 18 in Tables 2 and 3 for all velocities and are shown graphically in Figs. 5 and 6 for stack velocities of 500 and 1100 fpm.

Space limitations permit only a brief summary of observations made to determine discharge angles. Results indicate that the discharge angle decreases with increase in face opening for stack heads Nos. 1, 2 and 3. Discharge angles are less for a square throat as indicated in Nos. 1 and 4 than in a rounded throat stack head. With higher velocities Nos. 4 and 5 showed a lower angle of discharge. Stack head No. 6 in the same series does not conform due to the greater face to throat dimension. With a change from a 3-in. to 6-in. throat radius only a slight lowering of discharge angle can be noted in Nos. 1 and 7. Stack heads Nos. 8, 9 and 10, equipped with splitters, show a discharge angle about half of that for corresponding stack heads Nos. 1, 2 and 7. A comparison of Nos. 15 and 17 equipped with cushion chambers with those of Nos. 11 and 16, which are plain, show no appreciable effect on discharge angle. Heads Nos. 11, 16 and 18 which are square turns indicate an increase in discharge angle from 20 to over 30 deg with increase in face height from 6 in. to 12 in. Rounded back stack heads Nos. 4 and 5 indicate an increase in discharge angle of 15 deg with change in face opening from 6 in. to 9 in. as compared to half that increase for the square turns for Nos. 11 and 16. Turning vanes added to the square stack heads Nos. 20 and 21 reduced the angle of discharge to slightly below the horizontal.

The static pressure at the base of the stack heads at the various stack velocities is given in Column 19 of Tables 2 and 3. The results are also shown graphically for various groups of stack heads in Figs. 7 to 12 inclusive. Column 31 of Tables 2 and 3 shows this static pressure at the base of the stack head, which is also the static pressure drop to the atmosphere, in per cent of the velocity head at the base of the stack head.

A comparison of the static pressure drops for five heads with 14-in. x 6-in. outlets is shown in Fig. 7. The highest pressure drop, 0.3 in. at 1400 fpm, was found with No. 4, a short radius turn with a square throat. Fig. 8 shows the effect on the static pressure drop with the addition of turning vanes and splitters to the plain stack heads Nos. 1, 7 and 11, all with 14-in. x 6-in. face opening. In every case a reduction was experienced.
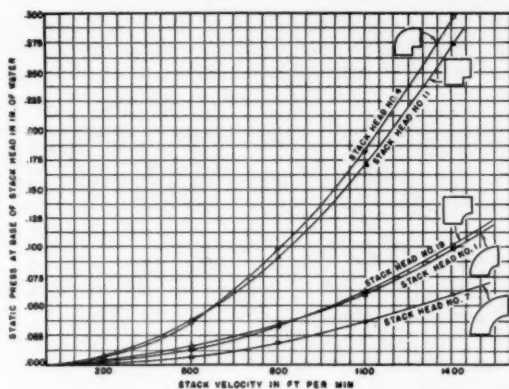
FIG. 7. STATIC PRESSURE AT BASE OF STACK HEAD FOR VARIOUS NON-EXPANDING STACK HEADS

FIG. 8. EFFECT OF TURNING DEVICES ON STATIC PRESSURE AT BASE OF STACK HEADS OF NON-EXPANDING TYPE
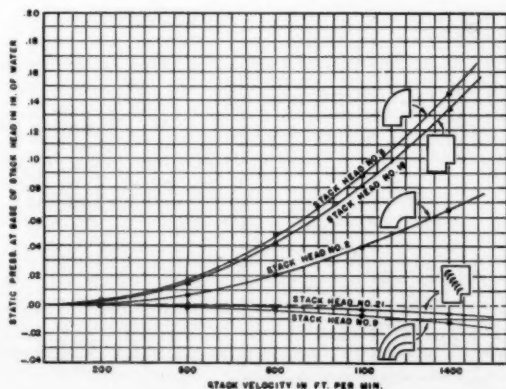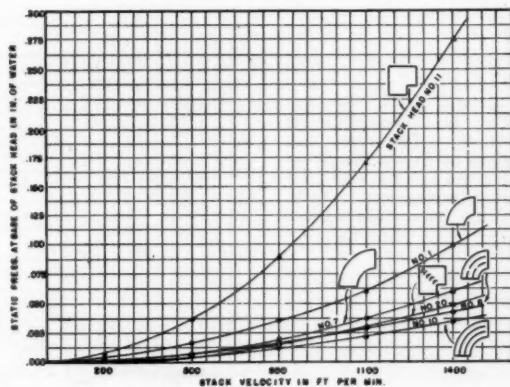
FIG. 9. EFFECT OF TURNING DEVICES ON STATIC PRESSURE AT BASE OF STACK HEADS OF EXPANDING TYPE
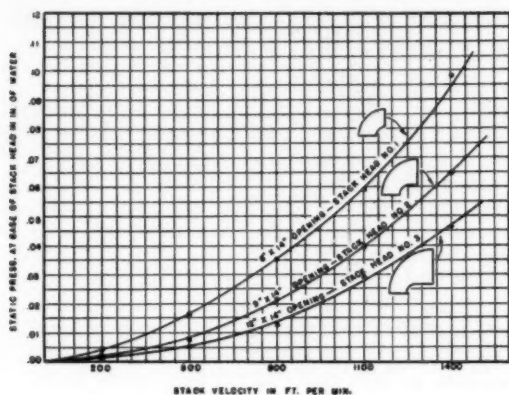
FIG. 10. VARIATION OF STATIC PRESSURE AT BASE OF STACK HEADS OF ROUNDED TYPE WITH THE FACE OPENING

FIG. 11. VARIATION OF STATIC PRESSURE AT BASE OF STACK HEADS OF SQUARE-THROATED ROUNDED-BACKED TYPE WITH THE FACE OPENING
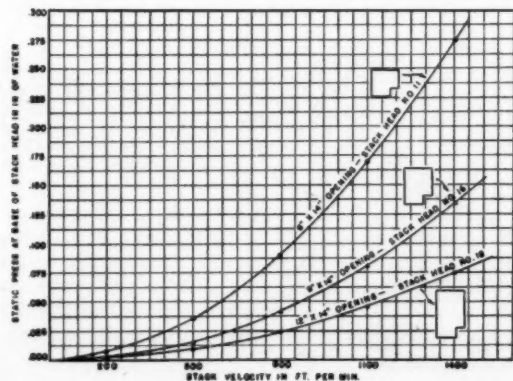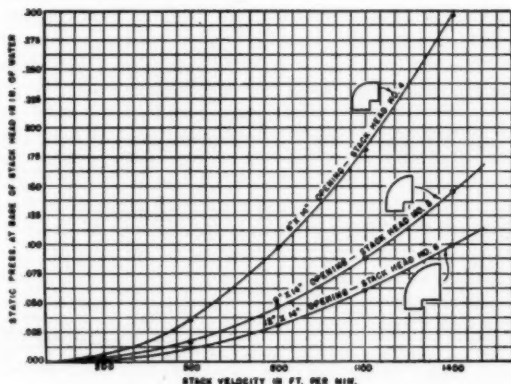
FIG. 12. VARIATION OF STATIC PRESSURE AT BASE OF STACK HEADS OF SQUARE TYPE WITH THE FACE OPENING

Expanding stack heads with a 14-in. x 9-in. face opening are shown on Fig. 9. The same relative positions as to drop in static pressure are found for rounded and square backed heads Nos. 5 and 16 as for the two with 14-in. x 6-in. face opening, Nos. 4 and 11.

The change in static pressure at the base of the stack head with increase in face opening is shown in Fig. 10 for short radius heads, in Fig. 11 for square-throated, rounded backed heads, and in Fig. 12 for square heads. In each case, an increase in face opening resulted in a lower static pressure at the base of the stack.

In all of these stack head tests the total energy of the air stream at the base of the head is used up in resistance through the turn and in resistance in being delivered out into the room air. The algebraic sum of the static and velocity pressures at the base of the stack head gives a measure of this energy. The addition of these energies at 1400 fpm for the short radius turns, Nos. 1 and 8, indicates that the splitters reduce the energy to 74 per cent. With the longer radius turns, Nos. 7 and 10, the reduction is only to 90 per cent, and with the expanding heads, Nos. 2 and 9, the addition of splitters reduced the energy to 58 per cent. Similarly for the square type stack heads with 14-in. x 6-in. face opening, Nos. 11 and 20, turning vanes reduced the energy to 42 per cent. In the case of the square type with a 14-in. x 9-in. face opening, Nos. 16 and 21, the energy was reduced to 44 per cent by the addition of turning vanes.

CONCLUSIONS

In most stack heads whether of the square or rounded type the air left at angles varying from 20 to 35 deg with the horizontal. The angle with the horizontal was less for stack heads with square throats than for those with an inner radius. A change from a 3-in. to a 6-in. inner radius only lowered the angle slightly. A greater face to throat dimension lowered the discharge angle. A larger face opening height resulted in a greater angle of discharge. This increase with an increase in opening height from 6 in. to 9 in. was twice as large with a rounded backed stack head as with a square backed stack head.

In most stack heads an area of reverse flow was found in the lower section of the face opening. The throat or inner radius was very important in determining the extent of this reverse flow, being largest with a square throat, less with a 3-in. inner radius and negligibly small with a 6-in. inner radius. The outer radius had practically no influence on this reverse flow area. The area of reverse flow increased almost directly with the face opening height with a given shape of stack head.

Large reverse flow area in general was associated with high angle of discharge and a large variation in face velocities.

Cushion chambers placed above square type stack heads had no effect on the distribution characteristics. The use of two splitters in the round type stack heads lowered the discharge angle with the horizontal to about one-half that of the plain fitting, made the face velocity quite uniform, and caused the reverse flow area to disappear or to become very small. The characteristics

were changed to those of longer radius turns. Little difference was found in the results when applied to the short or long radius turn. The use of five or six turning vanes with a 1-in. outer radius and a 2-in. inner radius resulted in good distribution as to direction, uniformity of face velocity and absence of reverse flow area.

The highest pressure drop was found with the square-throated rounded-back type. The square type showed only slightly less drop. On either of these the substitution of a 3-in. inner radius for the square throat caused the pressure drop to become very much less. The use of a 6-in. inner radius resulted in a further decrease in pressure drop. Splitters in stack heads with short radii and turning vanes in square stack heads lowered the pressure drop below that of the long radius stack head tested. Splitters in the long radius stack head reduced the pressure drop below that of the square and short radius types with turning devices.

When applied to expanding stack heads the turning devices resulted in lowering the static pressure at the base of the stack head below atmospheric pressure. This was due to the uniformly low velocity over the whole discharge area and the interchange of kinetic and pressure energies between the inlet and the outlet of the stack head.

The sum of the static pressure and the velocity pressure at the base of the stack head was considered to give a measure of the energy required to make the turn and to enter the atmosphere where all energy in these two forms is dissipated. In square type stack heads the addition of turning vanes dropped this energy to about 45 per cent, considering the energy required in the plain fitting as 100 per cent. The application of splitters to the long radius turn reduced the energy to 90 per cent and to 74 per cent with the short radius turn. With an expanding head this measure of energy with the application of splitters was reduced to 58 per cent.

The application of splitters and turning vanes improved the performance of stack heads materially.

DISCUSSION

H. O. CROFT (WRITTEN) : The authors are to be congratulated upon their undertaking a research problem which throws considerable light upon the flow characteristics and pressure losses of stack heads for ventilating purposes. Their findings, that the effects of *cushion chambers* (stack head No. 15) was negligible, is contrary to what one might first suspect but this is often the case when one relies upon pure speculation.

It would add considerably to technical knowledge if some of the experiments could be repeated with a piece of straight duct continuing horizontally from the stack head as there is some evidence to show that the elbow loss with the additional straight duct may be less than without the additional duct and, while such an installation might not be used for distribution in a room, the information obtained would be useful in estimating elbow losses.

The static pressure has been taken near the base of the head where peculiar flow characteristics, due to the change in direction, might influence the static pressure

recorded. Would it be feasible to repeat some of the experiments with the static pressure being taken about half way down on the stack at such a location that the possibility of influence, due to the change in direction of flow, would be minimized? Tests would then be run without the stack head to determine the loss in the straight vertical run which would, of course, be subtracted from the readings with the stack head.

It is hoped that the authors will continue their splendid work, using at least two other stack sizes. This additional information would furnish extremely useful experimental knowledge needed by engineers.

J. H. VAN ALSBURG: I think this probably should be clarified a little bit. This is one of a series of papers from a series of projects on air distribution. A previous paper [1] on instruments basically covered correlation and conclusions on the types of instruments to be used for definite work. It was necessary to do the instrument work before we could do the air distribution work. This paper is also a preliminary paper in two respects, namely, it is a very voluminous document that required a tremendous amount of work, and if you check the paper carefully you will readily realize a great number of tests were made.

With the additional papers on this same subject, we can define and determine the characteristics of air streams which in turn will have an application as to the air distribution and the measurement of air in a room.

JOHN JAMES (WRITTEN): Due to the turbulent action of air streams in duct systems it is particularly difficult to record the true directional effects of air as it leaves an outlet. The wood directional vane devised by the authors for measuring this effect seems to have accomplished the objective desired, as the results in Figs. 5 and 6 give excellent indications of the air velocity pattern for the various stack heads.

For a number of years some designers have expressed the opinion that stack heads similar to Nos. 15 and 17 in Fig. 6, arranged with a cushioning chamber would materially aid in providing a horizontal air flow pattern. The results reported in these tests indicate that this cushioning chamber has no effect on straightening the air flow as it leaves the outlet.

Another interesting observation from the standpoint of design, is that whereas it might be thought that stack heads with an inner radius would give a lower angle of flow with the horizontal; it was found that a square throat gave better results.

D. W. NELSON (WRITTEN): At the velocities used there seemed to be no serious disturbance at the base of the stack head influencing the static pressures. The values obtained were checked as to correctness with various stack head and velocity combinations by traverses of a static tube at that location and at lower levels in the stack as suggested by Professor Croft. Part of the static pressure measured is due to elbow loss and part is shock loss of the air entering the atmosphere. The shock loss form discharge from a stack head varies widely with the degree of uniformity of discharge and with the amount of expansion. This complicates the method suggested, of taking readings with and without the stack head.

Additional tests could well be made placing a piece of straight duct beyond the stack head but this particular study was limited to stack heads and was not to include elbows. At the time of this paper a few unreported tests have been made

[1] ASHVE RESEARCH REPORT No. 1140.—The Use of Air-Velocity Meters, by G. L. Tuve, D. K. Wright, Jr. and L. J. Seigel. (ASHVE TRANSACTIONS, Vol. 45, 1939, p. 645.)

on a smaller stack size. Considerable experience on instrument technique was carried over from the former study on air distribution from side wall outlets and considerable of the present experience will assist in the next study on the performance of side outlets on horizontal ducts. The present work will continue with the application of grilles to the faces of stack heads. A very large number of tests are involved and the problems must necessarily be limited to a reasonable scope.

SEMI-ANNUAL MEETING, 1940
Washington, D. C.

THE Society's Semi-Annual Meeting 1940 attracted a large attendance at the Wardman Park Hotel, Washington, D. C., June 17-19, when 341 members, ladies and guests were registered. Prior to the official opening of the meeting, the Council, the Committee on Research, and many technical advisory committees held meetings. Throughout the 3-day sessions there were many special committee meetings, so that the business and technical affairs of the Society were reviewed and plans were approved for new research projects, future meetings and many other activities.

Pres. F. E. Giesecke called the meeting to order on Monday, June 17, in the Continental Room and a brief message of greeting was given by T. H. Urdahl, general chairman of Committee on Arrangements of the Washington D. C. Chapter. He expressed the pleasure of the Washington members for the splendid attendance and extended the hospitality of the Washington members to the visiting members and guests. President Giesecke responded briefly on behalf of the Officers and members of the Society.

President Giesecke announced that H. H. Erickson would present the discussions on the report of the Committee to Study the Method of Selecting Society Officers and Council Members. The report had been printed and distributed to members in advance of the meeting. Mr. Erickson briefly reviewed the results of the Committee's work and then read written comments on the report from several chapters, among which were, Atlanta, Illinois, Oklahoma, Pittsburgh and St. Louis, and he also read letters from E. K. Campbell, F. C. McIntosh and W. C. Randall.

N. D. Adams, Rochester, Minn., said that the ideas contained in the various letters made it evident that members of chapters were not favorable to the recommendations of the Committee. W. H. Driscoll, New York, stated that a summary of the comments indicated that the local chapters showed a misconception of the purpose for which the Society was formed. He pointed out that it is a national body rather than a federation of chapters and, unlike the United States of America in which each state must have equal and specific representation, chapters of the Society have no such privilege or function The business of the Society is conducted at its Annual and Semi-Annual Meetings, and sometimes no chapter representative attends, although in his opinion it would be a fine thing if every chapter could be represented in the work of the Society and its influence felt in national affairs. When the chapters were given a voice in the management of the Society through Nominating Committee representation, it was evident that their conception of the pur-

pose of the Society and their understanding of the processes of the Society could have created a situation that might be detrimental to its best interest. He believed that the Committee could have recommended a better plan and thought that the special committee which made the study and rendered its report should review the situation and attempt to reconcile the thoughts of the members as a national organization rather than the ideas of the chapters.

C. H. Randolph presented the objections of the Wisconsin Chapter to the Committee's report. E. K. Campbell stated that he had served as chairman of the Nominating Committee and one of the difficulties encountered was the requirement for two meetings which sometimes changed the personnel of the group which met in summer and that which had attended the winter meeting. He suggested that the work of the Nominating Committee be completed at a single meeting. F. J. Pratt, Bremerton, Wash., presented the views of the Pacific Northwest Chapter, which felt that the Committee's suggestion for District E involved too large a territory. Prof. L. G. Miller, East Lansing, stated that the Western Michigan Chapter had discussed the Committee's report at length, and felt that the nominating method now in use was satisfactory.

On motion of N. D. Adams, properly seconded, it was proposed that:

A Committee consisting of H. H. Erickson, chairman, Messrs. R. H. Carpenter, W. A. Russell and three additional members, Tom Brown, W. H. Driscoll and F. C. McIntosh, appointed by the President of the Society, be requested to review the Committee's report and consider all of the comments submitted and present further recommendations to the Society at the 47th Annual Meeting in Kansas City. The motion was unanimously carried.

Messages of greeting were received from Allen W. Williams, managing director of *National Warm Air Heating and Air Conditioning Association,* and from Col. W. A. Danielson, former president of the Washington D. C. Chapter and now stationed at the Panama Canal Zone. A personal word of greeting was given by J. C. Fitts, secretary, *Heating, Piping and Air Conditioning Contractors National Association,* and J. F. Collins, Jr., secretary, *National District Heating Association,* brought a message of greeting from the members of his organization.

President Giesecke introduced C. A. McKinney, president of South Texas Chapter, who invited the members to attend the Fall Meeting of the Society scheduled for October 14-15 at Houston, Tex., and A. D. Marston, president of the Kansas City Chapter, who extended a cordial invitation to all to attend the 47th Annual Meeting to be held at Kansas City, January 27-29, 1941.

RESOLUTIONS

J. F. Collins, Jr., Pittsburgh, chairman of Committee on Resolutions, presented resolutions of approval, which were unanimously adopted on motion of W. R. Rhoton, Cleveland, and W. L. Fleisher.

WHEREAS, the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS has met in the City of Washington, D. C., for the Semi-Annual Meeting of 1940, and

WHEREAS, the Washington, D. C., Chapter of the Society and their ladies have in many ways and at all times shown their gracious hospitality, and

WHEREAS, the many committees of this Chapter have performed their many functions in an outstanding manner, and

WHEREAS, the Aerothermal Players, the Kensington Quartet and Stan Brown and his band have presented very fine and unusual entertainment and music, and

WHEREAS, the Banquet Committee arranged an evening of great pleasure for those who attended,

WHEREAS, the authors of the many valuable and educational papers have presented these to the benefit of those in attendance,

WHEREAS, the newspapers and trade press have given this meeting excellent publicity.

WHEREAS, the Greater National Capitol Committee of the Washington Board of Trade has given excellent cooperation and assistance,

WHEREAS, the accommodations and service of the Wardman Park Hotel have been unusually satisfactory,

WHEREAS, the officials and employees of the Federal Bureau of Investigation, the National Bureau of Standards, the Naval Testing Basin and Capitol Heating, Air Conditioning and Power Plants enabled this Society to inspect, in an unusual and very interesting way, the operations of these bureaus and buildings, and finally,

WHEREAS, the Washington, D. C., Chapter has lived up to its promise of a year ago that we should have cool and delightful weather throughout our visit to Washington,

THEREFORE BE IT RESOLVED, that this Society in meeting assembled express (to these various groups) its wholehearted thanks and sincere appreciation for the many courtesies, services and kindnesses which were enjoyed by all who attended this Semi-Annual Meeting, 1940.

GEORGE W. TUTTLE and JOHN F. COLLINS, JR.
COMMITTEE ON RESOLUTIONS

President Giesecke called for any new or unfinished business, and there being no further business, declared the Semi-Annual Meeting adjourned.

ENTERTAINMENT

Members and guests who arrived in Washington early on Sunday were greeted by Col. H. H. Downes and 24 enjoyed a day of fishing on Chesapeake Bay. Reports on the number of fish caught have not been classified up to this time.

On Sunday evening Mrs. T. H. Urdahl entertained at her home the wives of Officers, members of the Council and Committee on Research, including Mmes. Giesecke, Fleisher, Blankin, Collins, Walker, Stacey, Van Alsburg and Geren.

On Monday the Congressional Country Club provided the hazards for 32 golfers who participated in the tournament for the Research Cup and the Eichberg Memorial Cup. J. C. Matchett, Chicago, won the Research tournament and the Illinois Chapter team captured the Eichberg Cup.

Research Cup won by J. C. Matchett—Gross Score 98. Handicap 30—Net 68
Eichberg Cup won by Illinois Chapter—Net Scores by J. C. Matchett 68—C. E. Price 71—R. T. Miller 71

A group of 60 journeyed to the Laboratories of the National Bureau of Standards and were conducted through the various laboratories by R. S. Dill and others of the staff.

In the morning a group of ladies visited the White House and in the afternoon enjoyed a sightseeing tour of downtown buildings and some visited the Senate Gallery.

At 9:00 p.m. an informal party and show was held in the Continental Room of the Wardman Park Hotel under the direction of I. M. Day. Dancing was enjoyed after the show and a buffet supper was served.

A group of songs was given by the Kensington Village Singers, including Messrs. Willis Fiske, Lyall Peterson, Paul Johnson and Arthur Applegate.

The *Washington-Round Table* was conducted by I. M. Day and the cast included Clyde Hall, T. H. Urdahl, and E. J. Febrey. *Mis-Information Please* was supplied by Messrs. Febrey, Spurney and Urdahl, in reply to questions asked by Mr. Day.

In the original sketch *Going With the Wind,* the cast of Aerothermal players was: F. E. Spurney as *Whett* Butler, Lucille Dixon as *Rosie* O'Hara Butler, F. B. Sale as *Chic* Sale of the Perforated Utilities Co., F. A. Leser as B. G. Blowhard (high pressure salesman from Chicago), and P. H. Loughran, Jr., as Henry Zephr (timid salesman from Timbucktoo). The scene took place on the front porch of *Tarry* Hall, and was sometime between 1865 and the present.

In this sketch a unique piece of apparatus was introduced and space does not permit a complete description, but it is safe to say that the influence of this unit on Washington temperatures was so great that immediately following the Society's meeting temperatures dropped into the 50's and required the use of fireplaces and heating plants for three days.

In presenting this super-colossal unit, the high pressure salesman explained that it contained a blower and a compressor, and elaborate control features. In describing the control devices he went from bottom to top; lower right, *coldstat;* center, *comfort-stat;* top, *heat-stat;* left, *not-so-hot-stat;* right, *not-so-cold-stat.* On left side of unit he explained is the *fiddle-stat* which controls the base vibrations of the compressor. The *evaporative sprinkle unit* on top turns water into steam when the machine gets hot, also is useful for putting out fires. The *smoke pipe* on the right is to permit the escape of gases in case the motor or blower burn up, and it is provided with an *animal-stat* to take the bite out of fumes by uniform diffusion. The grille in front is a *perforated utilities,* dual purpose, indicating flow grille which provides accurate information as to the direction and velocity of the wind. The body of the unit is a fibrous product constructed of material which has withstood the test of time. It is produced by Packing, Box, and Caster Co.

On Tuesday morning the ladies visited Arlington, Mount Vernon, Lee Mansion and other points of interest on a motor tour and boat ride on the Potomac River. Many returned to witness the demonstration of the FBI at the Department of Justice Building. In the afternoon a group of ladies enjoyed a card party at the Wardman Park Hotel, and Mrs. Walter Heibel, Old Greenwich, Conn., and Mrs. M. Noble, Syracuse, N. Y., were the winners.

At 7:30 p.m. the Semi-Annual Banquet was held at the Wardman Park with M. T. Firestone acting as toastmaster. D. S. Boyden, past president of the Society presented a memory book to past president J. F. McIntire. A. E. Stacey, Jr., presented the Research Cup and M. W. Bishop, Chicago, presented the Society with a new cup to perpetuate the W. Roy Eichberg Memorial Cup, originally presented by the Philadelphia Chapter. The cup was accepted for the Society by M. F. Blankin, Philadelphia. A brief address was made by Hon. Luther Patrick, representative of the 9th District of the State of Alabama. After dinner dancing was enjoyed until after midnight.

Following the technical session on Wednesday inspection trips were organized to the Naval Testing Basin, Carderock, Md., Central Heating Plant, and the Capitol Power Plant and Air Conditioning Systems.

SEMI-ANNUAL MEETING PROGRAM

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

WARDMAN PARK HOTEL, WASHINGTON, D. C.
(All Events on Eastern Standard Time)

TECHNICAL SESSIONS

Monday, June 17

8:30 A.M. Registration—Main Lobby
10:00 A.M. *Technical Session*
   Summer Cooling Load as Affected by Heat Gain Through Dry, Sprinkled and Water Covered Roofs, by F. C. Houghten, H. T. Olson and Carl Gutberlet
   Air Conditioning Design of New Federal Office Buildings, by W. A. Brown
   Dirt Patterns on Walls, by R. A. Nielsen
   Discussion of Report on Nominating Committee changes

Tuesday, June 18

10:00 A.M. *Technical Session*
   Study of Changes in Temperature and Water Vapor Content of Respired Air in the Nasal Cavity, by L. E. Seeley
   Summer Cooling Requirements of 745 Clerks in an Office Building, by Dr. W. J. McConnell and M. Spiegelman
   Fever Therapy Locally Induced by Conditioned Air, by Dr. M. B. Ferderber, F. C. Houghten and Carl Gutberlet

Wednesday, June 19

10:00 A.M. *Technical Session*
   Methods of Rating the Noise from Air Conditioning Equipment, by John S. Parkinson
   Air Flow Through Supply and Exhaust Openings in Buildings, by G. L. Tuve and D. K. Wright, Jr.
   Effect of Room Dimensions on the Performance of Direct Radiators and Convectors, by A. P. Kratz, M. K. Fahnestock, E. L. Broderick and S. Sachs

ENTERTAINMENT EVENTS

Sunday, June 16

8:30 A.M. Fishing Party on Chesapeake Bay ($2.50 per person)

Monday, June 17

10:30 A.M. Ladies' Visit to White House (Bus leaves Hotel)
1:30 P.M. Golf Tournament—Congressional Country Club (Greens Fee $2.50)
   Members desiring to participate in tournament must register with Golf Committee at registration desk prior to 11:30 a.m. First bus leaves hotel at 12:30
2:00 P.M. Tour of Inspection—Bureau of Standards
2:00 P.M. Sightseeing Tour of Downtown Buildings
   Capitol, Pan American Union, Library of Congress, Supreme Court Building, Botanical Gardens, Washington Monument. Fare $1.30 per person (no charge for members' wives and children)
9:00 P.M. Informal Party and Show—Continental Room
   An original playlet, "Going with the Wind," written and presented by members of the Washington, D. C. Chapter—Dancing after the Show and Buffet Supper (Smorgasbord). $1.40 per person.

Tuesday, June 18

9:30 A.M. Ladies' Motor Tour and Boat Ride
> Lincoln Memorial, Lee Mansion, Arlington, Mount Vernon; Christ Church, Masonic Lodge in Alexandria; $2.50 per person (no charge for members' wives and children). Luncheon on boat. Bus will meet boat for return to Wardman Park or trip to FBI demonstration at 2:30 p.m.

1:30 P.M. Gold—Congressional County Club (Greens Fee $2.00 per person)
2:30 P.M. Tour through Federal Bureau of Investigation, U. S. Department of Justice (Round trip bus fare 30c per person)
2:30 P.M. Ladies' Card Party
7:00 P.M. Semi-Annual Banquet and Dance
> M. T. Firestone, Toastmaster—Presentation of Past President's Memory Book to J. F. McIntire—Address by Hon. Luther Patrick, Representative of 9th District, Alabama—Presentation of Golf Trophies

Wednesday, June 19

10:30 A.M. Style Show for Ladies—Shopping Tour
1:30 P.M. Inspection Trips to:
> (a) Naval Testing Basin, Carderock, Md.
> (b) Central Heating Plant
> (c) Capitol Power Plant and Air Conditioning Systems

COMMITTEE MEETINGS

Sunday, June 16

10:00 A.M. Meeting of Research Executive Committee—Hamilton Room
1:30 P.M. Council Meeting—Madison Suite
8:00 P.M. Meeting of Committee on Research—Madison Suite

Monday, June 17

9:00 A.M. Advisory Committee on Air Distribution and Air Friction—Hamilton Room
2:00 P.M. Advisory Committee on Radiation and Comfort—Adams Room
5:30 P.M. Guide Publication Committee Meeting—Hamilton Room

Tuesday, June 18

1:00 P.M. Advisory Committee on Insulation—Franklin Room
2:00 P.M. Advisory Committee on Air Conditioning in Industry—100-F
4:00 P.M. Meeting of Nominating Committee—Madison Suite
4:00 P.M. Advisory Committee on Heat Requirements of Buildings—Hamilton Room
4:00 P.M. Advisory Committee on Heat Transfer of Finned Tubes with Forced Air Circulation—Adams Room

Wednesday, June 19

2:00 P.M. Advisory Committee on Sound Control—Hamilton Room
2:00 P.M. Advisory Committee on Corrosion—Franklin Room
4:00 P.M. Advisory Committee on Cooling Load in Summer Air Conditioning—Adams Room

COMMITTEE ON ARRANGEMENTS

T. H. URDAHL, *General Chairman*

Entertainment—I. M. Day
Fishing—Col. H. H. Downes
Finance—G. R. Walz
Banquet—F. E. Spurney
Golf—W. E. Kingswell
Sightseeing—P. H. Erisman, Jr.

Technical Sessions—R. K. Thulman
Inspection Trips—S. P. Eagleton
Transportation—D. B. Tuxhorn
Registration-Reception—S. L. Gregg
Publicity—M. T. Firestone
Attendance—W. H. Littleford

Ladies—P. H. Loughran, Jr.

SUMMER COOLING LOAD AS AFFECTED BY HEAT GAIN THROUGH DRY, SPRINKLED AND WATER COVERED ROOFS

By F. C. Houghten,[*] H. T. Olson,[**] and Carl Gutberlet,[***] Pittsburgh, Pa.

A S part of the general research project of determining the cooling load for summer cooling and air conditioning, the 18 ft x 18 ft x 8 ft high cubicle shown in Figs. 1 and 2 was built by the ASHVE Research Laboratory during the spring of 1939. It was located on the roof of the Warehouse Building of the Pittsburgh Experiment Station of the U. S. Bureau of Mines, where it was exposed to full solar radiation from shortly after sunrise until just before sunset. The cubicle was built as a part of the program of the Technical Advisory Committee on Cooling Load in Summer Air Conditioning, consisting of C. M. Ashley, *Chairman,* John Everetts, Jr., M. G. Kershaw, A. E. Knapp, C. S. Leopold, L. S. Morse, R. M. Stikeleather, and J. H. Walker. This Committee is interested in heat gain through various types and orientation of building constructions, as affected by both the resistance to heat flow and heat capacity of the structures.

The test building walls included four 5½ ft wide x 7 ft high wall sections of 13 in. brick and plaster, and four of 4 in. brick, 8 in. hollow tile and plaster, each type of construction having north, east, south, and west exposures; also, three similarly sized panels of 4 in. brick veneer on frame construction consisting of wood sheathing, 2 in. x 4 in. studding, metal lath and plaster, with east, south and west exposures.

The deck of the horizontal roof of the test cubicle was divided into 5-ft square panels, consisting of one 4-in. gypsum, one 2-in. gypsum, one 4-in. tile, one 6-in. concrete, three 2-in. concrete, and two 2-in. yellow pine plank constructions, all covered with tarred felt and coal tar pitch roofing. One each of the 2-in. concrete and 2-in. pine panels were built so that the roofing over them could be either sprinkled or flooded with up to 6 in. of water. With one exception the built-up roofing was covered with approximately ½ in. of slag; this exception being the third 2-in. concrete panel which was left with a smooth asphalt finish during part of the summer and later with painted aluminum.

This paper deals with variations in the heat flow through a roof as affected

Fig. 1.   Test Cubicle Located on Roof of Warehouse Bldg. of
the Pittsburgh Experiment Station, U. S. Bureau of Mines

by the surface finish and by sprinkling or flooding during the summer heat
when the space below is cooled and air conditioned.

The inside of the cubicle was summer cooled and air conditioned to 75 F
and 50 per cent relative humidity throughout the study.   A Nicholls heat flow
meter was attached to the inside center of each of the side wall and roof
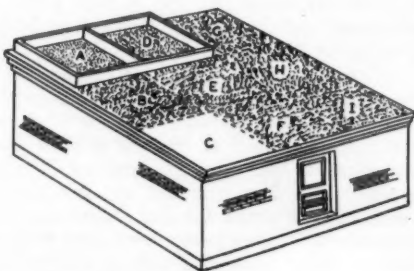
A-2" CONCRETE DOUBLE SLAG WATER COOLED PANEL
B-2" DRY SLAG CONCRETE PANEL
C-2" CONCRETE SMOOTH FLAT ASPHALT
D-2" PLANK DOUBLE SLAG WATER COOLED PANEL
E-2" DRY SLAG PLANK PANEL
F-4" DRY SLAG TILE PANEL
G-2" DRY SLAG GYPSUM PANEL
H-4" DRY SLAG GYPSUM PANEL
I-6" DRY SLAG CONCRETE PANEL

Fig. 2.   Sketch of Test House Showing Location of
Roof Panels

panels in order to give an hourly measurement of the rate of heat flow through their inside surfaces. Thermocouples gave the outside air temperature, the outside surface temperatures, the inside surface temperatures, and the inside air temperature 6 in. from the center of the panels, and, with the flooded roofs, the temperature of the water at different depths from the surface. Additional thermocouples gave the temperature part-way through some of the types of construction. The means for measuring the heat flow through the inside surface and for observing temperatures is shown in Fig. 3.

Figs. 4 to 8 give curves for heat flow through the inside surface, and temperatures through the roof panels from the air 6 in. below the inside

FIG. 3. INTERIOR OF TEST CUBICLE SHOWING MEANS USED FOR MEASURING HEAT FLOW THROUGH INTERIOR SURFACE AND MEASURING TEMPERATURES

surface to the air 6 in. above the outside surface throughout a 24-hour period, for the 2-in. concrete construction flooded with 6 in. of water, the 2-in. pine construction flooded with 6 in. of water, the 2-in. concrete with dry, smooth asphalt finish, respectively, all for the same day, August 30, 1939. Fig. 8 also shows the solar radiation intensity and wind velocity for August 30.

The heat flow through the concrete panel with the dry slag finish for August 30 and September 8 is given in Fig. 9, together with that through the 2-in. concrete panel covered with 6 in. of water for August 30, and through the concrete with the top surface damp or sprinkled for September 8. Similar heat flows for the 2-in. plank with the same top surface conditions are given at the top of the chart. The solar radiation, outside dry-bulb and wet-bulb temperatures are also given for the two days as well as the wind velocity for September 8.
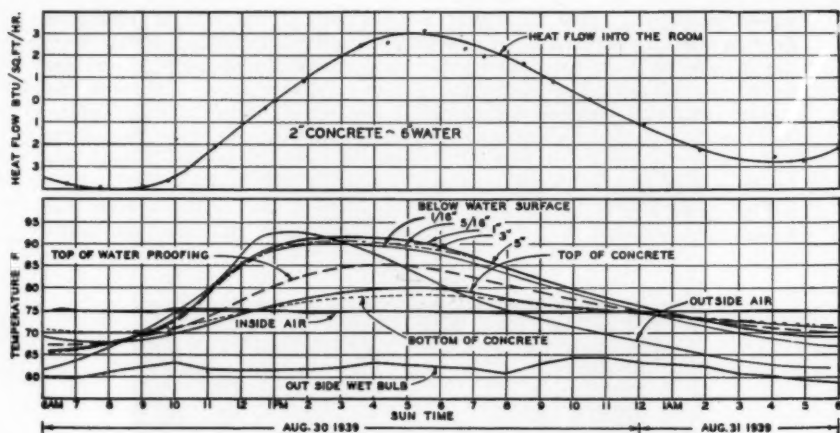
Fig. 4.   Relation between Time and Heat Flow through Inside Surface of 2-in. Concrete Panel Flooded with 6 in. of Water and Temperatures of Water and Other Points through Roof

The heat flows through the 2-in. concrete panel with the dry slag finish for September 2 and September 8, through the concrete panel with the damp surface for September 8, and through the concrete flooded with 1 in. of water for September 2 are given in Fig. 10. Similar heat flows for the 2-in. pine panel with like surface conditions are given at the top of the chart, while
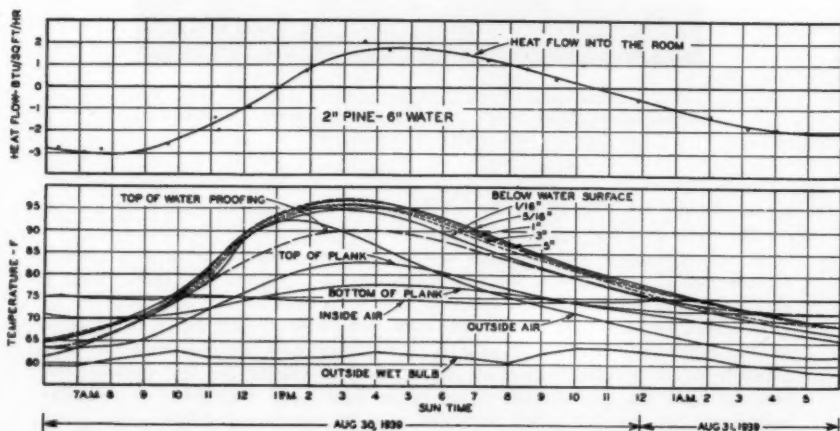
Fig. 5.   Relation between Time and Heat Flow through Inside Surface of 2-in. Pine Panel Flooded with 6 in. of Water and Temperatures of Water and Other Points through Roof

the solar radiation, outside dry-bulb and wet-bulb temperatures for the two
days are also given together with the wind velocity for September 2.

The heat flow through the 2-in. concrete panel with the smooth asphalt
finish on September 8 is given in Fig. 11, as well as those through the dry
slag finish on September 8 and September 15, and through the smooth asphalt
finish painted aluminum for September 15. At the center of the chart are
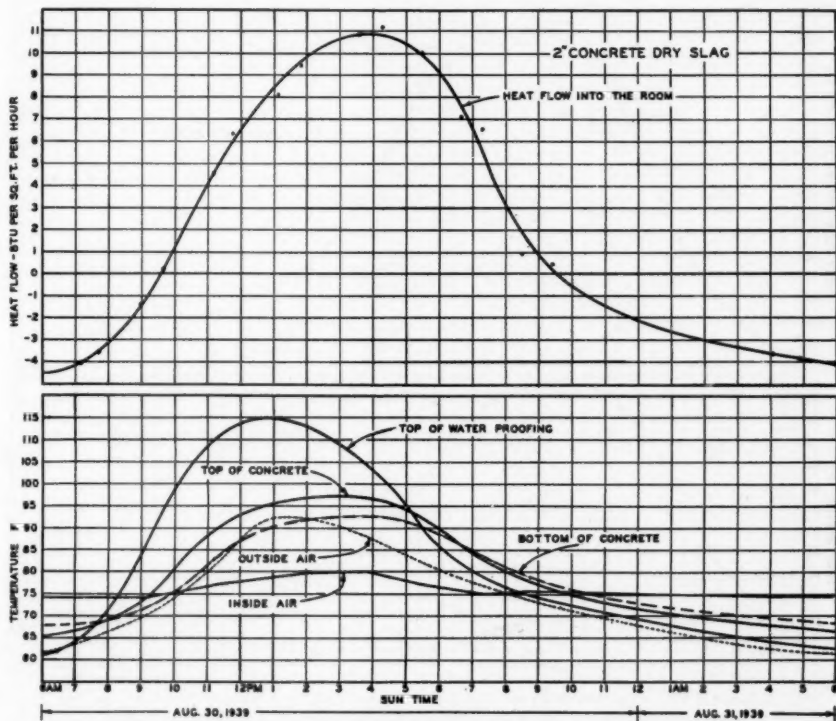
FIG. 6. RELATION BETWEEN TIME AND HEAT FLOW THROUGH INSIDE SURFACE OF
2-IN. CONCRETE PANEL AND TEMPERATURES NEAR AND THROUGH PANEL

shown the outside dry- and wet-bulb temperatures for the two dates, the top
surface temperatures of the smooth asphalt finish concrete panel, and the
temperatures for this same panel painted with both a varnish base aluminum
paint, and with a bituminous aluminum paint, for Sept. 15. The solar radiation
is given at the top of the chart for September 8 and 15 and the wind velocity
is given for September 15.

The curves in Figs. 4, 5, 6 and 7, and the curve for the concrete panel

with the smooth asphalt surface for August 30 in Fig. 8 give a direct comparison between the heat flow through the 2-in. concrete panel with 6 in. of water, with a dry slag surface, and with a smooth asphalt finish; also for the 2-in. plank panel with 6 in. of water and the dry slag surface.

The curves in Figs. 8, 9, 10 and 11 give comparisons of the heat flows through the concrete panel with the smooth black and smooth aluminum painted
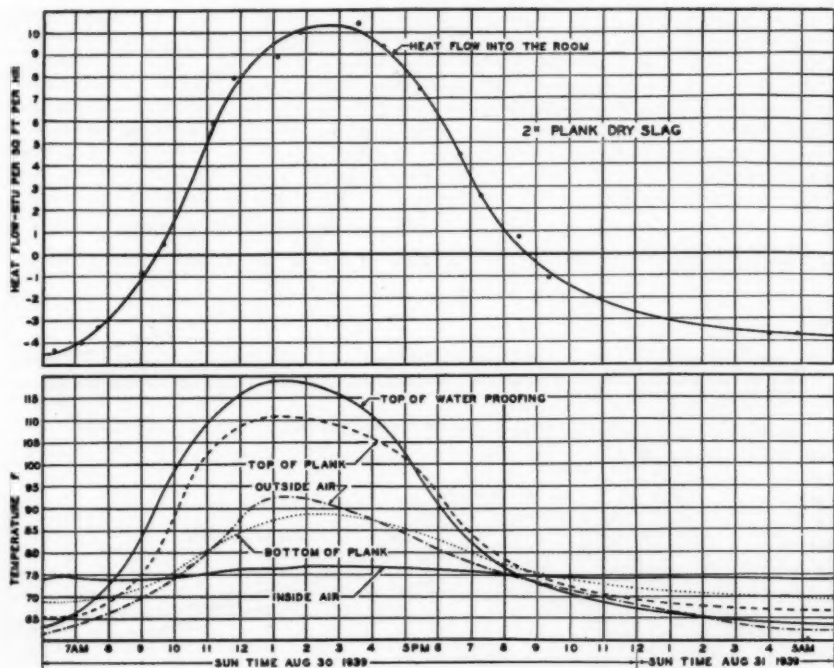
FIG. 7. RELATION BETWEEN TIME AND HEAT FLOW THROUGH INSIDE SURFACE OF 2-IN. DRY SLAG PINE PANEL AND TEMPERATURES NEAR AND THROUGH PANEL

asphalt surface, with the dry slag surface, with 6 in. of water, with 1 in. of water, and with damp slag; also for the 2-in. plank panel with dry slag surface, with 6 in. of water, with 1 in. of water and with damp slag, for different days.

The fact that only one concrete and one pine panel were available for 6 in. and 1 in. flooding and for sprinkling made it impossible to obtain a direct comparison between these different conditions on the same day. An attempt is made in Fig. 9 to adjust the data for the 6-in. flooded panel from August 30 to September 8; in Fig. 10 to adjust the data for 1-in. flooded panel from

September 2 to September 8; and in Fig. 11 to adjust the data for the smooth aluminum-painted asphalt surface from September 15 to September 8. This is done by plotting in each case the heat flow through the dry slag covered panels for the two days in question and by assuming that approxi-
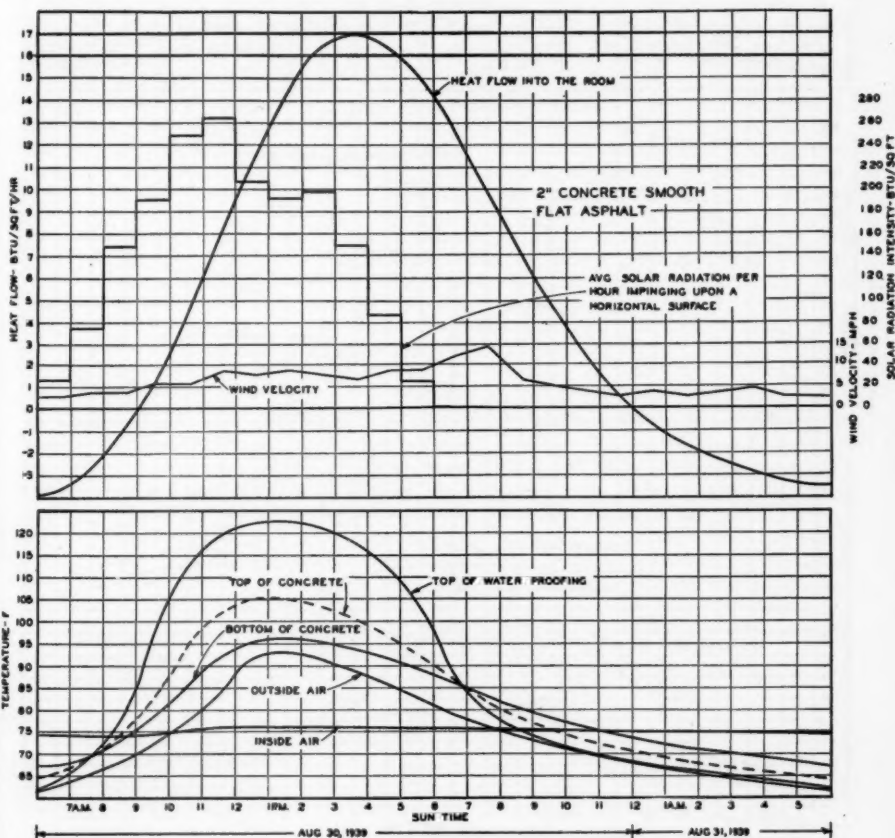
Fig. 8. Relation between Time and Heat Flow through Inside Surface of 2-in. Concrete Panel with Smooth Asphalt Finish, Also Relation between Time and Solar Radiation Impinging Upon Roof

mately the same percentage change would occur in the heat flow through the panel in question which was tested on only one of these two days. As a result of this assumption, heat flow curves through all of these panels are plotted as adjusted to a single day, September 8. While there is not sufficient background to place complete confidence in this adjustment the two facts,—
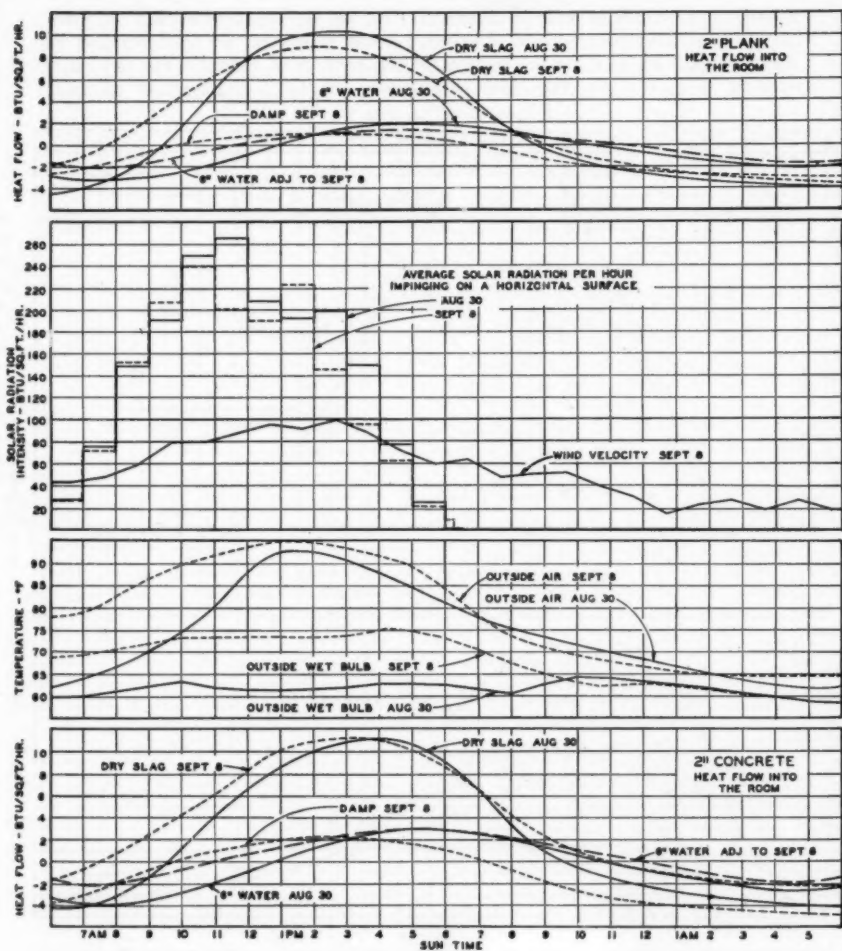
Fig. 9. Relation between Time and Heat Flow through Inside Surface of 2-in. Concrete Panel and 2-in. Pine Panel Dry, Flooded with 6 in. of Water and Damp; Also for the 6 in. of Water Adjusted to Sept. 8. The Solar Radiation Impinging Upon Roof and Wet- and Dry-Bulb Temperatures are Also Given
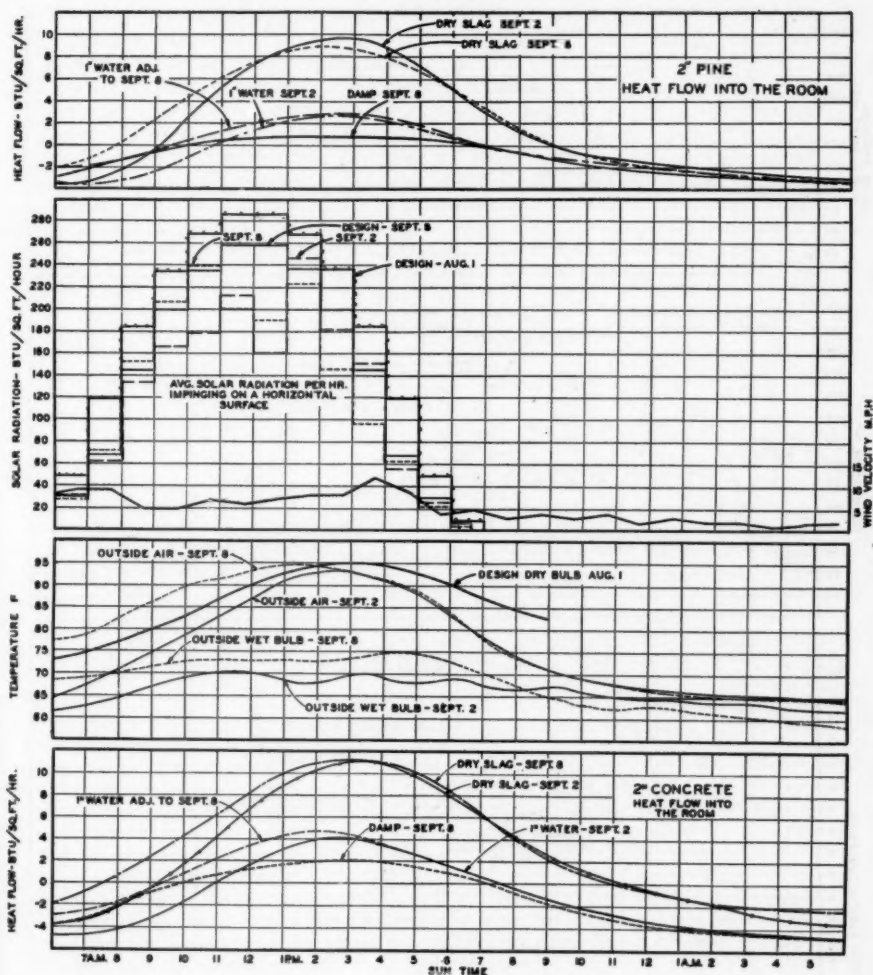
Fig. 10.   Relation between Time and Heat Flow through Inside Surface of 2-in. Concrete Panel and 2-in. Plank Panel, Dry, Flooded with 1 in. Water and Damp; Also for 1 in. of Water Adjusted to Sept. 8. The Solar Radiation Impinging Upon the Roof for Sept. 2 and 8, the Design Solar Radiation for Aug. 1 and Sept. 8, the Design Outside Temperature for a 95 F Day and Dry- and Wet-Bulb Temperature for Sept. 2 and 8 are Also Given
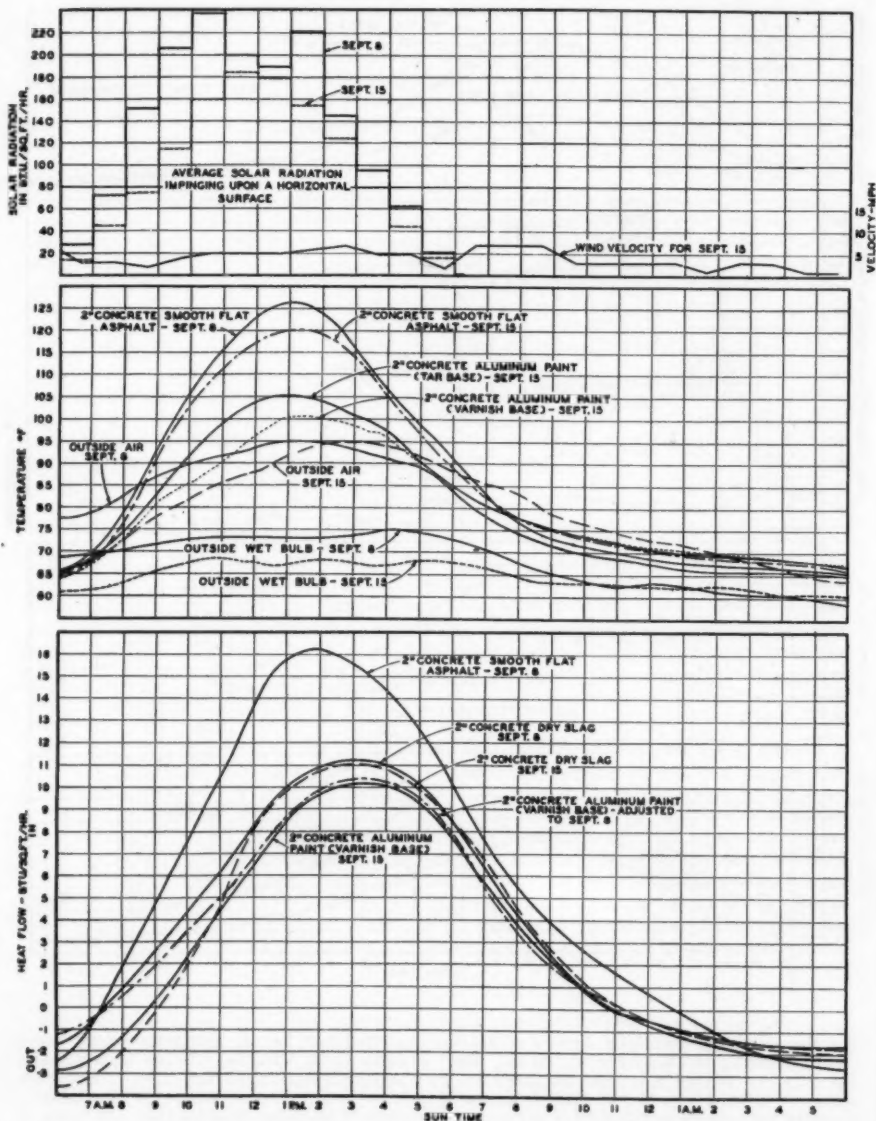
FIG. 11. RELATION BETWEEN TIME AND HEAT FLOW THROUGH INSIDE SURFACE OF SLAG CON-
CRETE PANEL FOR SEPT. 8 AND 15, THE 2-IN. CONCRETE PANEL WITH SMOOTH BLACK TAR
PITCH FINISH FOR SEPT. 8 AND THE 2-IN. CONCRETE BLACK TOP PANEL PAINTED ALUMINUM
FOR SEPT. 15 AND ADJUSTED TO SEPT. 8. THE SOLAR RADIATION IMPINGING UPON ROOF FOR
SEPT. 8 AND SEPT. 15, THE SURFACE TEMPERATURES AND DRY- AND WET-BULB TEMPERATURES
FOR SEPT. 8 AND 15 ARE ALSO GIVEN

first, that some adjustment in this direction must naturally apply, and *second,* that the adjustment between the two days is never great, should lend considerable confidence to the comparison.

The series of heat flow curves given in Fig. 12 as reproduced from the charts, Figs. 4 to 11, gives a direct comparison between the heat flow through the different panels with different surface conditions as observed for, or adjusted to, September 8. The same series of heat flow curves, adjusted to September 8, and given in Fig. 12 are repeated in Fig. 13, adjusted to what has
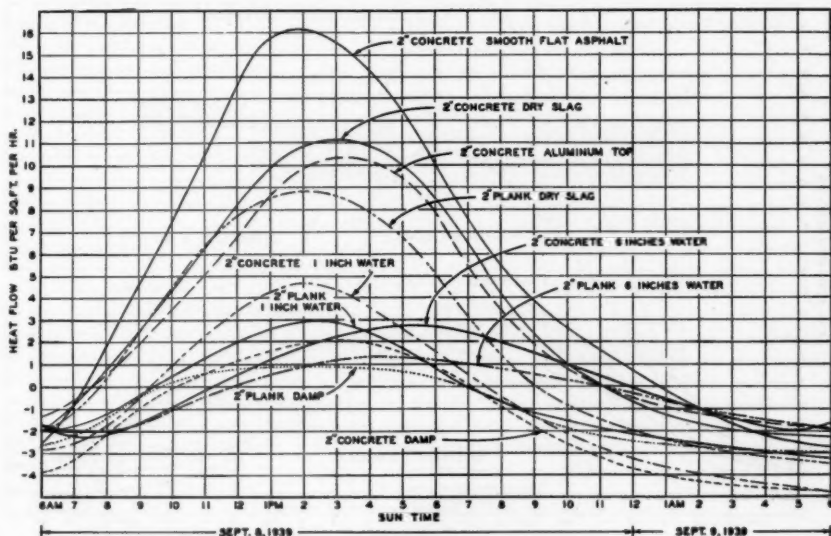
FIG. 12. RELATION BETWEEN TIME AND HEAT FLOW THROUGH INSIDE SURFACE OF SEVERAL HORIZONTAL ROOFS STUDIED, ALL DETERMINED FOR OR ADJUSTED TO SEPT. 8

been assumed to be the design outside dry-bulb temperature and the design solar radiation intensity on a horizontal surface for August 1.

It is of interest to note the considerable reduction in heat flow through either the damp or flooded concrete roofs below that given for the same roofs with dry slag or with smooth asphalt without and with aluminum painted surfaces. Approximately the same relationship is shown for the 2-in. pine panel; however, all of the heat flow values are somewhat smaller for the wood panels than for the concrete panels. A comparison of the 6-in. flooded, the 1-in. flooded, and the damp roofs all corrected for the same day, September 8, in Figs. 8 and 9, shows maximum heat flows of 2.8, 4.7 and 2.1 Btu per square foot per hour, respectively. Of greater interest, however, is the fact that these reductions are from a value of approximately 11 Btu per square foot per hour for the dry roof of the same construction, with the exception that,

the water proofing membrane on the damp and flooded roof was a little thicker as shown in the description, Fig. 14.

The effect of water in either the case of the sprinkled or of the flooded roof is to greatly reduce the rate of heat flow from that found for the same panels in dry condition. Of greater interest, however, is the effect of the water to absorb a large part of the radiant heat, to retain it with a uniform temperature throughout the water depth, and to dissipate it back to the air through the latent heat of evaporation. It will be seen that during maximum
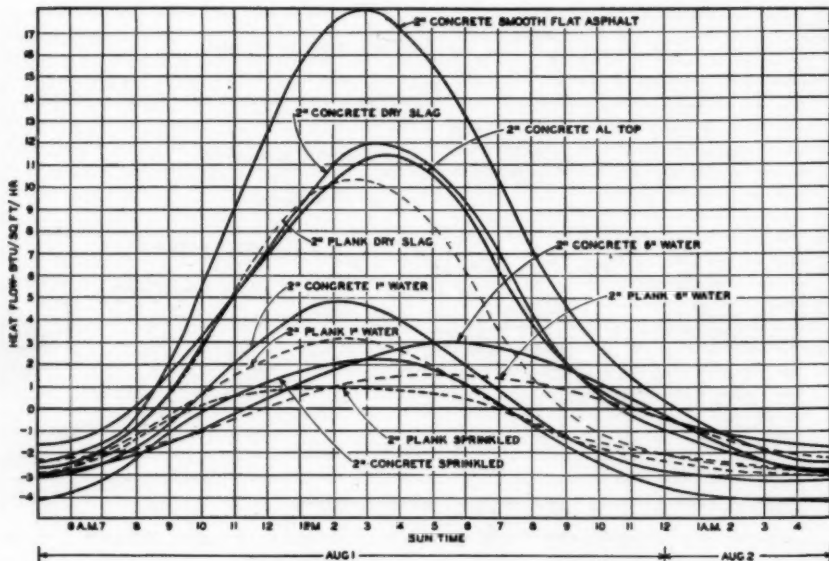
FIG. 13.   RELATION BETWEEN TIME AND HEAT FLOW THROUGH INSIDE SURFACES OF SEVERAL HORIZONTAL ROOFS STUDIED, CORRECTED TO DESIGN DAY, AUG. 1

temperature conditions the total temperature variation through the depth of the water never exceeds 4 deg and that this temperature spread is usually considerably less. There is also interest in the fact that during the time that solar radiation is effective the lowest temperature is found 1/16 in., or immediately below the top surface of the water. This obviously results from the lowering of the top surface temperature by evaporation while a considerable part of the radiant energy is transformed into heat through the depth of water with a somewhat elevated water temperature at the 5 in. depth above that found at other points. The elevation of the temperature at the bottom of the water obviously must result from greater absorption of solar radiation at the level of the top surface of slag.

The comparison between flooding with 6 in. of water and just maintaining

the slag damp, Fig. 9, shows that the 6 in. depth of water serves to reduce the rate of heat flow through the panel during the early part of the day and to effect a somewhat greater rate of heat flow thereafter. This is obviously due to the lag in the rise in water temperature during the early part of the day and the lag in the drop in this same temperature during the latter part of the day and the night due to the heat capacity of the water. The one inch of water, having a lower heat capacity, warms up more rapidly, gives a higher maximum rate of heat flow earlier in the day than is the case for the 6-in. flooded roof.

The fact that the aluminum painted surface considerably reduces the heat flow through an otherwise black roof is amply demonstrated in earlier Labora-
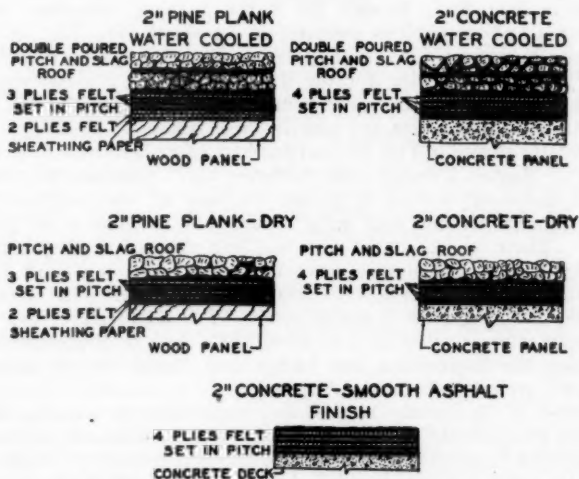
FIG. 14. TYPICAL CONSTRUCTION OF ROOFS TESTED

tory publications.[1, 2] This is again brought out in Fig. 11 for the 2-in. concrete with a smooth asphalt finish, without and with aluminum paint. This comparison indicates a maximum rate of heat flow through the aluminum painted, smooth asphalt panel of 10.3 Btu per square foot per hour on September 8. This surface was painted with a varnish base aluminum paint. On the same day a very small section of the same panel was covered near the edge where it would not affect the heat flow meter reading, with a tar base aluminum paint giving the difference in surface temperature indicated in the center of Fig. 11. It will be noted that the varnish base aluminum gave a reduction in the top surface temperature, and therefore in the heat flow,

[1] ASHVE RESEARCH REPORT No. 853—Absorption of Solar Radiation in Its Relation to the Temperature, Color, Angle and Other Characteristics of the Absorbing Surface, by F. C. Houghten and Carl Gutberlet. (ASHVE TRANSACTIONS, Vol. 36, 1930, p. 137.)
[2] Symposium on Thermal Insulating Materials, March 8, 1939. (*American Society for Testing Materials*, Phila., Pa.)

below that given for the bituminous aluminum paint.   No heat flow rates were determined for the bituminous aluminum painted surface.

As compared with the maximum heat flow rate of 11.2 Btu per square foot per hour for the dry slag covered roof on September 8, Figs. 9 and 10, the black asphalt finish roof on the same day gave a maximum heat flow rate of 16.2 Btu per square foot per hour.   It should be pointed out, however, that in the case of the slag covered roof the slag pebbles were placed on the tar pitch giving additional thickness and heat capacity, so that the indicated difference in heat flow represents the combined effect of the difference in color and thickness of the slag roof.   The cross-sections of these roofs are shown in the sketches, Fig. 14.

It is of interest to note the considerable drop in temperature, indicating resistance to heat flow, through the water proofing consisting of several layers of felt and tar pitch as indicated in the sketch, Fig. 13.   Thus, at the time of maximum heat flow the temperature drop through the water proofing was 5, 7½, 7 and 7½ deg for the flooded concrete, dry concrete, flooded pine, and dry pine, compared with temperature drops of 1½, 4½, 3½ and 21 deg through the concrete and pine in these respective panels.

There is also plotted in Fig. 10 what has heretofore been used by the Laboratory [3] as an August 1 design solar radiation day.   Although the solar radiation on a horizontal surface is greater on June 21, the combined effect of high air temperature and high solar radiation is assumed to be greatest on August 1.   There is also plotted what has been assumed by the Laboratory to be a design outside temperature curve for a 95 F day on August 1.   It will be observed that September 8, 1939 was low for a design solar radiation day, but not far from right for a design outside 95 F temperature day.   Actually the air temperature reached 95 F at about one o'clock on September 8; during the morning the temperature was higher and during the afternoon it was lower than a typical 95 F day on August 1.

The curves in Fig. 12 may be used for comparative purposes in considering the relative effectiveness of the various roof constructions and surface effects. The adjustment involved in correcting the curves proposed for design purposes in Fig. 13 from the various test days between August 30 and September 15 to design conditions of dry-bulb and solar radiation on August 1 was greater than was the case for Fig. 12.   The maximum outside dry-bulb temperatures on August 30, September 2, 8 and 15 were 92.6, 93.3, 94.5, and 95.0 F, respectively, while the maximum solar radiation intensity on a horizontal surface on these same days was 265, 246, 239 and 185 Btu per square foot per hour.   The maximum design outside dry-bulb temperature for August 1 should be 95 F, while the maximum design solar intensity on a horizontal surface assumed for this date is 285 Btu per square foot per hour.   It will be seen that practically design temperature conditions were realized on September 8 and 15, with but slightly lower temperatures on the other two days. The solar radiation intensity on the test days was always lower than design for August 1.   This should naturally be the case since the solar radiation intensity on a horizontal surface is falling off rapidly during August and September; it being maximum on June 21 and decreasing from that date to a minimum on December 21.   Hence, while the adjustment from the other three

[3] ASHVE Research Report No. 1147—Heat Gain Through Glass Blocks by Solar Radiation and Transmittance, by F. C. Houghten, David Shore, H. T. Olson, and Burt Gunst. (ASHVE Transactions, Vol. 46, 1940, p. 83.)

test dates to September 8 on account of solar radiation was not very great, the adjustment from the test dates to August 1 was considerably larger.

However, a comparison of the location and magnitude of the crest for the different curves in Figs. 12 and 13 indicates that the adjustment to August 1 was never very large. Since there obviously must be some adjustment in the direction indicated, and since there was a fair basis for the assumption used in making this adjustment, it is believed that the curves in Fig. 13 are not greatly in error and that they can therefore be used with considerable confidence until additional work, which is proposed to be carried on in the same cubicle in the near future, makes available more satisfactory design data.

DISCUSSION

E. P. HECKEL (WRITTEN) : I was most pleased to receive a preprint of the paper which was presented at the Semi-Annual meeting of our Society at Washington, on the subject of summer cooling load as affected by heat gain through dry, sprinkled and water covered roofs. This represents, as usual, an outstanding job of research well done and with definite information that is of great importance.

Strange as it may seem, preparation for similar tests but on a much lesser scale had already been started here in Chicago by Samuel R. Lewis and myself, and a few observations had been noted when the above preprint came to my attention. Needless to say we immediately abated our efforts and accepted with appreciation the research conclusions.

The curves and test data of this paper indicate very definitely and clearly the advantages of roof sprinkling as compared to water covered roofs and that the heat flow in Btu's per sq ft per hour is more effectively reduced by sprinkling.

I am of the opinion that we can accept the data as developed by the investigation with very reasonable assurance of correctness. I do believe, however, that in actual practice the heat flow through any roof covered with from 2″ to 6″ of water would be found to be somewhat greater than that shown by the tests made, and particularly during a protracted period of high heat and sunlight intensity. The accumulation of soot, dust and oil fouling of water surfaces results in retardation of evaporation and cooling effect with resultant higher water mass temperature. I have observed higher water mass temperatures when water surface was fouled with an oily film and therefore readily assume that only clean, unfouled water was used in the tests from which the research data were developed.

This reference does not in any way affect the test conclusions of a sprinkled roof, for here we have continuous water flow with a roof-scrubbing effect, preventing any serious fouling of water even though recirculated from accumulator basin or from refrigeration machine condensers. This has in many instances of actual practice given very excellent results in reducing heat flow even though concrete data were lacking but which fortunately have now been made available by the excellent work of our research laboratory.

E. J. RODEE (WRITTEN) : This paper shows the insulating effect which may be obtained by flooding flat roofs in the summertime to diminish the cooling load of the building. The temperature curves and heat flow curves as shown in Figs. 4, 5, and 6, are very interesting and should be analyzed more fully by comparing the calculated heat flow through the various materials and under surfaces of the various roof constructions to the room air, using the temperatures given, with the measured heat flow.

Referring to Fig. 4 (2 in. concrete flooded with 6 in. water) the maximum heat flow occurs at about 5 P.M. and, as would be expected, the maximum temperatures through the roof are at this time. The heat flow as measured should approximate the heat flow as calculated. This does not appear to be the case. The measured heat flow is at the rate of 3 Btu per square foot per hour. The calculated heat flow from the under surface at a temperature of 78 F to room air at 75 F as shown by these temperature curves would be at the rate of 3.6 Btu per square foot per hour. When using a surface coefficient of 1.21 Btu per square foot per hour per degree temperature difference (heat flow downward) and 5 Btu per square foot per hour, if the heat flow is calculated from the top surface of the concrete under the waterproofing to the room air; coefficient of concrete 12 and temperature at top of concrete 80 F. A similar discrepancy occurs in both Figs. 5 and 6; the calculated heat flow is one and one-half to one and three-quarters as great as the measured amount.

As stated previously, the time of maximum heat flow would be expected to occur at the time of maximum temperatures through the roof and especially at the time when the difference between the surface temperature of the underside of the roof and the room air is greatest. Figs. 4, 5, 6, and 7, show this to be true but in Fig. 8 the maximum heat flow, as measured, occurs, between 3 and 4 P.M., while the time of maximum temperature difference between the under surface and the room air is about two hours earlier.

It would be interesting to determine the amount of insulation necessary to be installed in order to perform the same insulating effect as the water on the flooded roofs. It is quite possible this amount would not be excessive and in case this was found to be true, the permanent insulation would be more desirable than the temporary flooding system inasmuch as the permanent roof insulation would prevent heat loss in winter as well as heat gain in the summer.

J. H. WALKER (WRITTEN): This paper illustrates the viewpoint of the Research Laboratory, that the heat flow through roofs and walls in summer, under the complex cyclic conditions which exist, should be determined by actual measurement rather than by calculation. It is planned that this first series of studies on roofs will be followed by similar studies on walls of various types, with the ultimate aim of securing enough data so that the designing engineer will no longer be obliged to guess at this important component of his cooling load.

It seems apparent that heat gain through a roof or wall cannot be adequately expressed by a single constant such as we use in estimating wintertime heat loss but must be expressed by time curves such a those which the authors present. The broad program which the Committee on Research has undertaken for the study of heat flow through roofs, walls, and windows, is aimed at producing such a set of time curves so that the designer, with the aid of such curves, can if he desires calculate the hour-by-hour cooling loads for each room of a building.

Because of the pronounced effect of sun angle, it will be necessary to translate these Pittsburgh results to other latitudes and that will involve certain mathematical assumptions and much laborious calculation. It would be highly desirable to have some of these actual measurements duplicated at some laboratory in a more southerly latitude. Such an undertaking would be most appropriate for a Chapter to undertake as a chapter research project.

DIRT PATTERNS ON WALLS

By R. A. NIELSEN,* EAST PITTSBURGH, PA.

INTRODUCTION

A T THE end of the heating season in the spring the majority of homes are not uniformly dirty, but the walls often show a mottled or striated appearance, as illustrated by Figs. 1, 2, and 3. These furnish a picture of the structure behind the wall, similar to an x-ray, silhouettes of studding, laths, nails, etc., that form the wall and ceiling structure behind the paint, wallpaper, and plaster, which become more prominent as the heating season progresses. It is the purpose of this article to discuss the processes that may be responsible for the production of the wall patterns and possible methods that may be used to minimize them.

DIFFUSION AND CONVECTION

If a quantity of dirt is suspended in the air, it will be observed that the larger particles settle out quickly while the smaller ones, which in still air would settle out if given sufficient time, are prevented from settling by turbulence and convection currents. Together with these, there are very small particles in the air which would not settle out by gravity even in still air. In their action these particles begin to resemble gas molecules, the more so the smaller they are. Since fractional micron particles are but slightly larger than a mean free path for gas molecules in ordinary air, they no longer remain motionless in the air but bob about as they are hit at random by these molecules. Such particles are said to undergo Brownian movement; the smaller they are, the more the unevenness of gas molecule collisions affects them and the more violent and rapid their motion becomes. Such particles go in irregular paths that zigzag about. The important point is that the particles do not stay in one place but travel slowly and at random. Thus, even in still air, a particle near a wall will wander about until perhaps soon or after a long time it hits the wall. Since smaller particles move faster than the larger ones, a greater number of the small ones hit the wall in a unit time. Thus dirt deposited by diffusion will consist of more small than large particles.

The speed of Brownian diffusion will be illustrated by an example. If 0.1μ (micron) dirt particles are placed 0.1 mm from a wall, they will slowly spread out in all directions. From formulas [1] it is calculated that it will take, on the average, 25 seconds for half the particles to reach the wall; for $1\ \mu$ particles, again 0.1 mm from the wall, it will take ten times as long. If the particles were started twice as far away from the wall, they would, on the average, require four times as long to get there. For the range of temperatures found in rooms, the Brownian diffusion is independent of temperature. From this it is concluded that the rates of diffusion of the particles are very slow and are significant for dirt precipitation, only if the particles are very

FIG. 1. TYPE OF CEILING PATTERN CHARACTERISTIC OF UNINSULATED CEILINGS; CLEANEST SECTIONS ARE BENEATH STUDDING; DIRTIEST BETWEEN LATHS. WALLPAPER CLEANER WILL COMPLETELY REMOVE PATTERN

near the walls. Diffusion will thus tend to deposit dirt uniformly on the walls, preference being given to the smaller particles.

As mentioned previously, convection and turbulent air currents also cause suspended particles to migrate; by these means particles are stirred and carried about rather quickly (a few feet per minute). The air currents decrease in velocity in close proximity to a wall, and there their magnitude can only be estimated. Very close to a wall, the velocity along the wall as well as any velocity perpendicular to it is assumed to be zero. Particles are thus continually being brought close to the wall by convection. Should convection bring them into contact with the wall, it would be expected that they should, as in the case of diffusion, be deposited fairly uniformly over large areas, and should not deposit so as to form local dark and light areas of such spacings as are used between laths. Any patterns formed by convection should indicate lines of air flow, not structure behind the wall surface.

Convection tends to keep dust from settling out of the air and, in this

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[1] The Kinetic Theory of Gases, by L. B. Loeb. (McGraw-Hill Book Co., 1934, p. 399.)

way, helps to maintain a higher dust concentration in rooms; however, it seems that the principal action of air currents in regard to structure patterns on room walls is to maintain the concentration of dirt particles up to within a small distance of the wall. This dirt, brought by convection, is the reservoir from which the dirt that deposits on the wall is taken. The actual transportation of the dirt, between the place where the velocity of the air currents
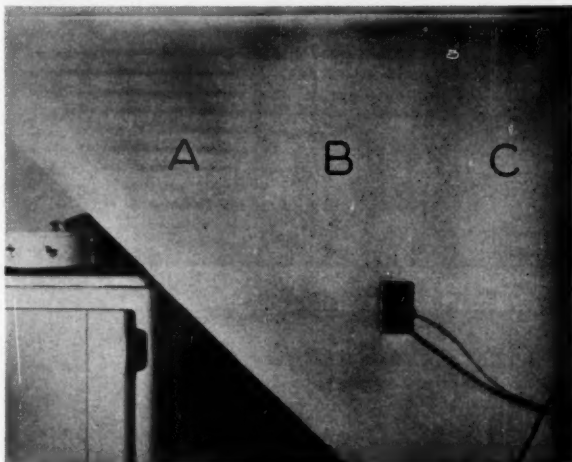
FIG. 2. A WALL PATTERN

Section *A* shows the characteristic lath marks; better insulation results in a more uniform Section *B*. The heat conductivity of the studding between *B* and *C*, as evidenced by the vertical dark area behind which is the studding. Note at the top of the picture the clean line at the wall junctions

becomes negligible and the wall surface, can be performed by diffusion and thermal precipitation.

THERMAL PRECIPITATION

Years ago, it was noticed by Tyndall, Rayleigh, Aitken, Lodge, Clark,[2] and others that heat had an effect on dust; a cold rod suspended in dusty air got dirty, while a hot rod remained clean. If a warm rod is put into a box filled with smoke, and suitably illuminated, a thin smoke-free region will be formed close to the rod's surface and the air rising directly above the rod will be clean. The thickness of the clean area depends on the temperature gradient near the rod.[3] Utilizing this effect, thermal precipitators have been built

[2] The work was reported in the literature between 1870 and 1885. References and partial abstracts of these works are given in references 3, 4 and 10 (see pp. 250 and 257 for references 4 and 10).
[3] The Dust-Free Space Surrounding Hot Bodies, by H. H. Watson. (*Transactions Faraday Society*, Vol. 32, 1936, p. 1073.)
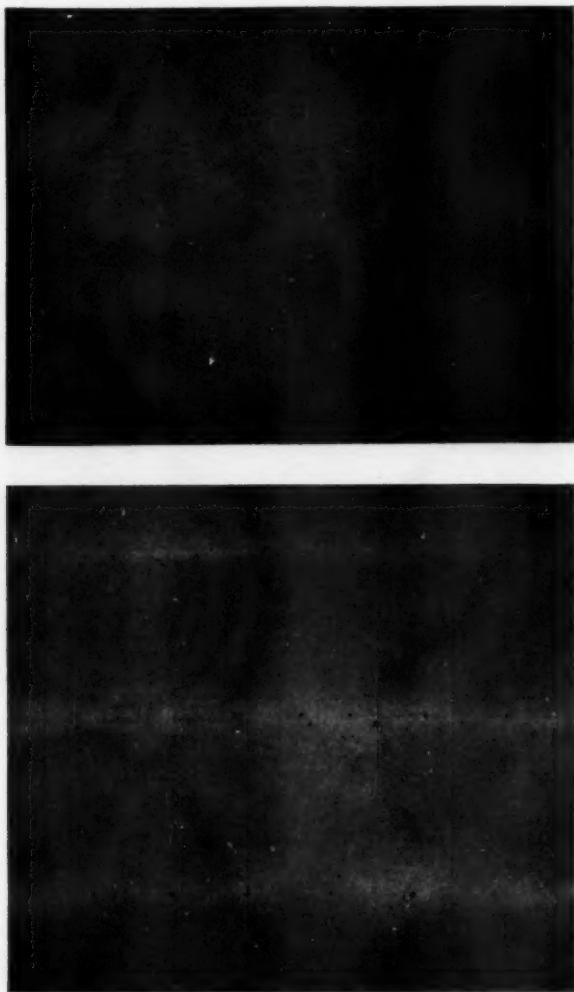
FIG. 3.   CEILING SECTIONS OF AN UNINSULATED PLASTERBOARD
CEILING.   *A* (*above*) IS ENAMELED; *B* (*below*) WALLPAPER-
COVERED.   NOTE THE LOCATION OF NAILS AND STUDDING

and found to operate satisfactorily.[3, 5]   The instruments take various forms

[3] See p. 249.
[4] Clouds and Smokes, by W. E. Gibbs. (Blakiston's Son & Co., Philadelphia, 1924, p. 63.)
[5] The Cleaning of Air and Gas by Thermal Repulsion, by S. C. Blacktin. (*Journal Society Chemical Industry*, Vol. 58, 1939, p. 334.)
Thermal Filters, by W. D. Bancroft. (*Journal Physical Chemistry*, Vol. 24, 1920, p. 435.)

but essentially are a hot wire in a cold tube or a hot wire between parallel plates. The plates or tube must be sufficiently close to the wire (an eighth inch or less, depending on the temperature gradient) that the clean area surrounding the hot wire extends as far as the cold walls. Then, air passing between wire and walls will be completely cleaned.

The action of a temperature gradient on dirt particles suspended in air must be considered. As previously mentioned, small particles in still air bob about due to their being hit by random numbers of gas molecules with various speeds

Fig. 4. Some Painted Pipes Near a Wall. Cold Water Pipe, Originally Similar to the Others, Has Become Very Dirty

and directions. If there is a temperature gradient, heat energy is being transmitted by gas molecules from the hot to the cold region; the heat energy is transmitted by the molecules as kinetic energy of their motion. With such a transfer of energy comes necessarily a transfer of momentum. The particle, suspended in a region in which a temperature gradient exists, receives more momentum on its side toward the hot region than on the side facing the colder one; thus it moves toward the colder surface. Perhaps the effect could be more simply explained by saying that a dirt particle is hit harder by the faster moving molecules from the hot surface than it is by the slower molecules which hit it on the side facing the cold surface; thus it moves down the temperature gradient. Einstein[6] derived a formula for the force on small

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[6] Zur Theorie der Radiometerkrafte, by A. Einstein. (*Zeitschrift Physik*, Vol. 27, 1924, p. 1. Loc. Cit. Note 1, p. 374.)

particles (less than 10^{-5} cm diameter) due to their being in a temperature gradient.

$$F = \frac{N}{2} KLS \frac{dT}{dx}$$

N = number of molecules per cubic centimeter
K = Boltzmann constant
L = mean free path of gas molecule
S = surface area of particle (proportional to r^2)
V = velocity of gas molecule
T = absolute temperature
$\frac{dT}{dx}$ = temperature gradient

The velocity of the dirt particle is given as $\frac{1}{8} V \frac{L}{T} \frac{dT}{dx}$.

From the last formula it will be noticed that for these very small particles the velocity is independent of the particle size. For particles above 0.1 μ the force becomes proportional to the radius instead of to the radius squared.[7] However, since the velocity of migration of the particle follows Stokes' law (velocity is proportional to force divided by particle radius), the velocity of these larger particles is also approximately independent of their radius since the force is now proportional to the radius. Therefore, the velocity of drift of particles in a thermal gradient is about the same for all particles.

Measurements of temperature gradients in rooms were made using a No. 36 (B & S gage) copper-constantan thermocouple, a type K potentiometer, and a sensitive galvanometer. The junction was supported away from the wall by taping the wires to the wall 1 in. or so from the junction. At the wall, open or covered thermocouple junctions normally gave a steady potential; when away from the wall as much as 1 mm, fluctuations of readings due to air currents were often observed. From the data of Watson[8] it is found that for a 9 F temperature difference between hot and cold surfaces, the clean region extends from the hot surface 0.1 or 0.2 mm toward the cold one, the distance being determined by convection. From this it is estimated that, in rooms, convection probably acts to within a fraction of a millimeter of the wall but at smaller distances the thermal gradients either form a protective dust-free layer over the surface, if that surface is warmer than the surrounding air, or they aid in depositing dirt on the wall if the air is the warmer. From temperature measurements taken between the wall surface and a point a centimeter or two from the surface, curves were obtained from which the temperature gradients at the surface were estimated. (Measurements very close to the wall are generally meaningless since most anything used in making the measurements disturbs the conditions.) The measurements herein, however, indicate the magnitude of the gradients that may be expected.

Two sets of readings taken about 10 min apart at the same point on the inside of a plastered and painted brick and tile wall (of a 12 x 15 x 24 ft laboratory room) illustrate the variableness of conditions and the influence of convection. Fig. 5 shows a steep gradient with wide temperature fluctuations

[7] Loc. Cit. Note 1, p. 378.
[8] Loc. Cit. Note 3.

caused by a warm air supply register on the opposite wall near the ceiling; the points on the curve represent average values of the temperature. The other curve was obtained after the fan feeding that register was turned off; here the readings were steady, especially near the walls. It was found that wall gradients were not constant but depended on many factors, in particular, convection.

A series of measurements in the third floor apartment of a house, of which only the ceilings and walls of the third (top) floor apartment were glass wool insulated, illustrates the magnitude of the temperature gradients that may be expected. The house is of brick veneer construction with slate roof and is heated by hot water radiators. With an inside temperature of 75 F and an
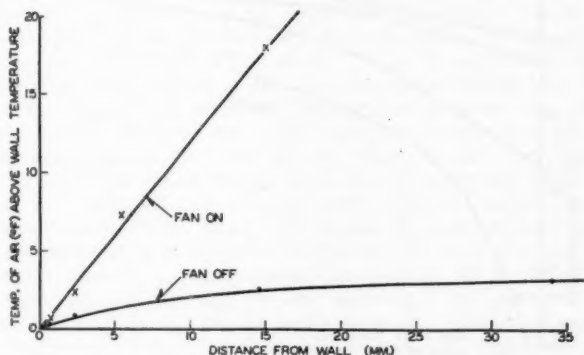
FIG. 5. WALL-AIR TEMPERATURE DIFFERENCES FOR A LABORATORY ROOM HEATED WITH HOT AIR

outside one of 30 F temperature gradients up to about 1 F per mm were found at the wall surface.

There are numerous interesting commonplace dirt patterns that any postulated mechanism of dirt deposition must explain. For example: before the third floor apartment was insulated with glass wool, the ceilings of the house became badly striated during the heating season and showed *light* regions below studding and under laths. Fig. 1 (though not of the same house) illustrates this type of pattern formation. Before the next heating season, the apartment was insulated as described, with the result that the patterns on the third floor walls were much different. The entire ceiling was quite clean and uniformly so, except for a series of *dark* rather diffused stripes below the studding. Without insulation, the greatest heat flow was through the plaster between laths (steep temperature gradient at the plaster surface); laths are poorer heat conductors than plaster, so the ceiling beneath the laths should appear lighter; similarly for the studding. After insulating, the low heat conductivity of the glass wool made the combined lath-plaster-glass wool section of the ceiling less conducting than the studding. Below the studding one might therefore expect the deposited dirt to be more dense than on other

ceiling sections and there should be no evidence of lath marks. This was exactly what was observed. On the insulated ceiling, the surface temperature below studding was 1 F cooler than between them.

This particular house has a gable in every room on the third floor. The sides of some of the gables gradually acquired an appearance as illustrated in Fig. 2. (Temperature measurements were made on a north bedroom wall but the painted kitchen wall was photographed since the design of the bedroom wallpaper interfered with good photographic reproduction of the lath marks.) The markings are interpreted as being due to irregular and skipped spots in
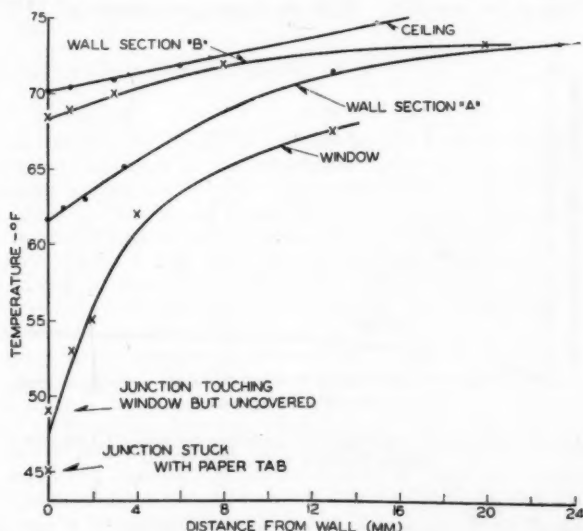
FIG. 6. AIR TEMPERATURES NEAR WALLS OF A NORTHERN EX-
POSURE ROOM

the insulation. In Section A the temperature of the surface in front of the lath was from 0.05 to 1 F warmer than surfaces between laths. Similar temperature differences between the clean and dirty stripes of lath marks were found on the second floor walls which are of the brick veneer type construction but without glass wool insulation.

In the bedroom the uniform section (corresponding to B of Fig. 2) was 6 F farmer than the striated section A. The clean parts of the striations of A are dirtier than any part of section B. Curves showing the temperature as a function of the distance from the wall at points A and B are shown in Fig. 6; the temperature gradient at A is larger than at B and extends a greater distance from the wall. An increase in the convective currents in the room, due to colder inside window temperatures or hotter radiators, will increase the gradient at A more than at B, since it will effectively equalize the air tem-

peratures near A and B at the same distance from the wall. Then, since A is colder with respect to the air temperature than is B, the temperature gradient there may be expected to be considerably higher than at B. No measurements were made during very cold weather, but it is common observation that the darkening of walls occurs much more rapidly at that time. This is ascribed to the increased heat loss and convection, both of which increase temperature gradients at the walls, as well as to the increased concentration of dirt in the air during that period.

In Fig. 3 are shown two pictures of the ceilings of a one-story uninsulated house with walls and ceilings of plasterboard. The ceiling shown in Fig. 3A is heavily enameled. The sections under the studding are lighter (cleaner) than neighboring ceiling parts. The presence of nails holding the plasterboard is indicated by prominent spots of much dirt deposition. On the left of Fig. 3A the joint in the plasterboard does not show, but its presence is indicated by the double row of nail marks. Near the right-hand edge there is a narrow but clean region at the junction of the ceiling and the inner wall. At the junction of walls the temperature gradients are necessarily reduced due to the geometry; also, since negligible air movement should exist there, one would expect very low thermal gradients and thus clean areas. Fig. 3B of the same house shows prominent perpendicular ridges due to lapping of the wallpaper; the less prominent ones are caused by lappings of underneath layers of the paper. As before, the studding and nails make known their locations, and this time the butting of two plaster-boards shows as a dark line between a double row of nail marks. When the ceilings are cleaned, there remains no sign of any structure. The patterned dirt deposit is on the surface and can be easily removed. The photographs show a fairly uniformly darkened ceiling in the case of either wallpaper or enamel-covered plasterboard. The studding are poorer conductors of heat than the plasterboard; the areas below them are warmer and thus the studding marks are relatively clean areas. Through the plasterboard and into the studding go fairly heavy nails which conduct heat from the wall and ceiling surfaces to cool points deep in the studding, consequently the nail heads are cooler than the surrounding areas and the temperature gradients near them are greater than at neighboring points. Dirt is thus deposited there faster than at nearby points and the nail heads *showing* through the walls and ceiling can be seen. Actual temperature gradients at various nearby points need not differ greatly in order to form structure patterns since small changes in the amount of deposited dirt, if in a regular pattern, can be seen very easily, especially if viewed obliquely.

Fig. 4 is included because it shows an outstanding case of thermal precipitation. The five yellow painted pipes along the wall carry gas, vacuum, compressed air, hot and cold water. Obviously the right-hand pipe carries the cold water; a small section was wiped off so as to show that underneath the dirt coating it was painted the same as the other pipes. The pipe was cold but there were no marks to indicate that water had at any time condensed on the pipe.

Another example of thermal precipitation is the familiar patterned walls near radiators and warm air registers. Above the radiator the wall is generally dark; especially is it noticeable just at the top of the radiator where light and

dark vertical patterns show the paths taken by air currents. The section of wall actually behind the radiator (excepting the top couple of inches) is really clean since it is heated by radiant energy to a temperature above that of the passing air. Convective and diffusive forces tend to deposit dirt on the wall but there the temperature gradients are large enough to *repel* all dirt and the wall remains clean. Above the radiator the wall temperature becomes lower than the warm air temperature; convection and thermal gradients then act together with the result that the wall soon becomes coated with precipitated dirt. The warm air rising to the ceiling establishes a high temperature gradient there and the result is that the ceiling becomes dark. Above a supply register the case is similar to that of the radiator. The heated air is carried close to the wall and the temperature gradient is sufficient to cause precipitation of some of the dirt. If a hood or baffle is placed over the top of the radiator, it is found that the wall above the radiator is cleaner because the warm air has been deflected away from the wall and cooler air has replaced it. The old saying that "hot air is dirty" is true in that hot air certainly has the ability to establish temperature gradients which indicate on the walls that the air was dirty. Present day air supply grilles are often fastened snugly to the baseboard and painted over. However, after a year or so the paint at the joint between baseboard and grille may crack or be cracked due to shrinkage, warpage, or other causes and a slit between the grille face and the baseboard results. When this has occurred, a black smudge a half inch or so in length soon forms above the crack. In this case thermal precipitation is again to blame. The heated air slowly escapes through the crack and passes across part of the baseboard; the low velocity and high temperature of the air form a good thermal precipitator which deposits dirt on the cooler baseboard where it is not wanted.

The inside surfaces of windows become dirty in a rather short time, even if they are kept well above the dew-point. For a single-glass window with air temperatures of 75 F inside and 30 F outside, temperature gradients were found to be about 5 F per millimeter persisting for 3 mm before starting to level off (see Fig. 6). The actual surface gradient was undoubtedly steeper but even so this was at least five times the gradient at the room walls and more than ten times the gradient at the ceiling.

In air at 70 F the velocity of particles in a thermal gradient of 1 F per millimeter is calculated as 1.3 in. per hour. Since such a gradient was found in a room in moderate weather under minimum convection currents, let it be assumed that such a gradient persists for 24 hours a day for 6 months. In other words, the dirt in 3.2 cu ft of air would be deposited on 1 sq in. of wall during that time. This is a conservative estimate, since in very cold weather not only are the temperature gradients large, but also the dirt concentration is high. This means that a large percentage of the dirt deposition and pattern formation that takes place during the heating season occurs during the, often brief, cold periods. From blackness tests made during the winter months in large industrial cities where soft coal is used, it is known that 3 cu ft of air drawn through 1 sq in. of filter will darken it appreciably, depending, of course, on the day. The calculations, though very approximate, indicate at least that thermal gradients are of the right order of magnitude to account for the observed blackness.

FILTRATION, ELECTROSTATIC CHARGES, AND HUMIDITY

There are other means by which it may be possible to deposit dirt on walls. One of these is filtration. If air is drawn through a filter, the dirt is strained out and the filter darkens. The same is true for a wall. With a pressure difference across a wall there will be a flow of air through it, sometimes in, sometimes out. If the flow is outward, a deposit of dirt on the interior wall should be observed. The magnitude of the leakage [9] depends on the kind of plaster, the care used in plastering, the surface finish, wind direction, and velocity. The leakage through a section of wall will not be uniform, but will depend upon the structure of the wall since lath, plaster, and studding offer various resistances to air flow. Thus it seems possible to produce lath patterns on walls and ceilings due to difference in the resistance to air flow offered by lath and plaster. However, if this mechanism explains the presence of the cleaner regions under studding, and the dark spaces between laths, it fails to explain their reversal, namely, dark regions under studding and the elimination of lath marks when glass wool is used as insulation. Other observers [10] report that the surface finish, "waxing, painting, enameling, and ordinary wall-papering, does not appear to affect the peculiar affinity of dust for certain regions on the wall, particularly those regions over the spaces between lath." Bonnell and Burridge report that lath patterns could be obtained even when a layer of lead foil was used.[11] In the case of the ceilings shown on Fig. 3, one would expect no filtration through nail heads, but some through plasterboard. Thus the filtration theory predicts white spots below nail heads! It must therefore be concluded that, though it is possible to produce a wall structure pattern by filtering air through a wall, it is not the method by which most homes acquire lath marks and other dirt patterns on their walls.

It is also possible for electrical forces to produce precipitation of dirt in a room. However, it will be shown that, as in the case of convection, it is improbable. It has been found experimentally that clean air blown through ducts acquires no appreciable charge, but if it is dirty, charges of considerable magnitude can be obtained.[12] Also, it is reported that air blown over plaster becomes negatively electrified.[13] Voltages may also be present due to space charge such as is normally found in outside air. However, it seems hopeless to try to make electrical forces account for (1) the elimination of lath patterns and the reversal of studding marks after a ceiling was insulated with glass wool, (2) to explain light regions under studding, and, (3) to explain the formation of lath marks even when lead foil covers the surface.

It has been suggested that moisture is responsible for the pattern formation. The influence of moisture can be divided into two cases: (1) the walls are transmitting moisture into the room, (2) the room is transmitting moisture

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[9] ASHVE RESEARCH REPORT No. 888—Air Leakage Through Various Forms of Building Construction, by F. C. Houghten, Carl Gutberlet and C. A. Herbert. (ASHVE TRANSACTIONS, Vol. 37, 1931, p. 177.)

[10] The Deposition of Dust on Walls, by W. H. Hooper. (*Physics*, Vol. 1, 1931, p. 61.)

[11] The Prevention of Pattern Staining of Plasters, by D. G. R. Bonnell and L. W. Burridge. (Department of Scientific and Industrial Research, England Building Research Bulletin, No. 10, February, 1931.)

[12] Generation of Static Electricity in Blower Systems, by A. H. Nuckolls. (Underwriters Laboratories Bulletin of Research No. 8, April, 1939.)

[13] Loc. Cit. Note 10.

to the walls. If the walls are transmitting moisture to the room, the effect is a bombardment by water vapor molecules leaving the surface upon any particle near the surface. Its action is to maintain the surface cleaner than similar dry sections.[14, 15] If the transmission of water vapor is in the reverse direction, one finds a force driving particles to the wall surface. Pattern formations are possible, therefore, if the various parts of the wall have different structures so as to permit the transmission of water or water vapor at different rates. Dampness of the surface may have some effect on the sticking of dirt which hits the wall; of this very little is known. However, experiments have shown that building a wall impervious to moisture or coating a wall surface with wax [16] does not prevent the formation of lath patterns. Arguments similar to those used against filtration and electrical forces will apply to humidity or moisture effects.

The effect of dampness and temperature gradients in keeping a surface clean is well illustrated in the case of the lungs. The action of constantly evaporating moisture and that of temperature gradients between the warm lung linings surrounding the cooler air help to keep dirt from depositing on lung tissues. As the hot rod when put into a smoky atmosphere surrounds itself with a dust-free space, so do the lungs establish a region about them through which gas molecules pass freely but dirt penetrates with difficulty.

CONCLUSION

Regardless of what method may be responsible for depositing dirt on the walls, that deposit can be minimized by reducing the dirt content of the air. This means air cleaning; the better the cleaning, the greater the reduction of the wall patterns. Air cleaners with high efficiencies are very useful for minimizing or even eliminating wall patterns as well as dirt deposits found near warm air supply registers, etc.

Though it is nice to have things clean, actually people will not object to a large amount of dirt on walls if it is deposited uniformly. A ceiling uniformly dirt-covered looks as before, except it has a grayer tinge. However, that same amount of dirt distributed non-uniformly, in lath patterns for example, is very noticeable, often objectionable. Since the amount of dirt deposited by thermal precipitation depends on the product of the dirt concentration by the temperature gradient, uniform heat insulation will reduce both general and patterned dirt precipitation on the walls by equalizing and minimizing temperature gradients.

Of the various methods by which it is possible to produce striated dirt deposits on walls, it appears that thermal gradients are responsible for the majority of those that are formed in homes during the heating season. The principal action of convection in pattern formation appears to be increasing the magnitude of the temperature gradients and in maintaining high dirt concentrations in the air by decreasing gravitational settling out.

[14] Loc. Cit. Note 3.
[15, 16] Loc. Cit. Note 10.

STUDY OF CHANGES IN THE TEMPERATURE AND WATER VAPOR CONTENT OF RESPIRED AIR IN THE NASAL CAVITY

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INTRODUCTION

THE project, being new, required the development of apparatus and procedures. Any experimenter will know what he is spared if an account of the development is omitted. Conceived in 1936, with preliminary work in 1937, it was not until 1938 and 1939 that what seemed like satisfactory results were obtained. The project was to measure changes in the temperature and water vapor content of respired air at certain positions in the nasal cavity with room air at various temperatures and relative humidities. The reason was a desire to discover if the control of relative humidity in occupied enclosed spaces was really important. Body heat loss studies were in general establishing the view that the condition of the air was important only as it enabled bodies to reject, without discomfort, an amount of heat equal to that evolved by body metabolism. Assuming no extremes in the humidity of air, this view is scientifically accurate for any static condition which achieves thermal equilibrium. Unfortunately, conditions are not in fact static because individuals are themselves mobile. They are going from place to place—indoors and outdoors—back and forth. They are literally passing almost instantly from one climate to another. Becoming acclimatized to daily and seasonal outdoor changes is sometimes in itself a severe process in temperate climates. Modern conditions involve subjection to changes which may induce consequences more important than is generally realized. Obviously, it is necessary to learn what actually happens to respired air in the nasal cavity before the effects of a sudden climatic change can be completely evaluated.

The following may help to unfold the problem. Cold air of moderate relative humidity will, upon being heated, have a much lower relative humidity. This happens when cold, outdoor air leaks into a heated but unhumidified structure. The cold air, upon being heated, is called *dry* and is believed by many persons to be unhealthful. Yet in heating that air the water vapor is not lost. Its

density per unit volume is only reduced somewhat because the volume of air is somewhat increased due to its rise in temperature. This means that, when breathed, the water vapor per unit volume is not much different whether the above air is warm or cold.

If the air in both cases (viz., warm and cold) is heated and humidified in the nasal cavity to the same final conditions, it follows that the moisture supplied in the nasal cavity would be about the same for equal volumes of air breathed. The so-called drying of the mucous membrane would be very little

Fig. 1. Partial View of Test Rooms

more in one case than in the other. However, if the air is not heated and humidified to the same final conditions, the moisture losses in the nasal cavity would be different. The sensations associated with the *dry* atmosphere might be a definite consequence of a higher rate of moisture loss especially when suddenly imposed. Whether or not an adequate adjustment can be made eventually to an altered atmosphere would depend upon individual characteristics. Passing from the atmosphere into a heated or cooled building involves an altered atmosphere.

When heat losses from the body by radiation and convection are not sufficient then the evaporative loss from skin surfaces makes up or attempts to make up the difference. Skin surfaces are sometimes moist and sometimes not.

The respiratory system is different. The surfaces therein are normally moist. These moist surfaces should be responsive to any and all changes in the water vapor pressure of inspired air regardless of whatever may be the extent of the evaporative heat loss from the external surfaces of the body. A body may be protected by clothing but the respiratory system, especially the nasal cavity, must cope with whatever air the environment provides. Persons

PER CENT RELATIVE HUMIDITY OF ROOM AIR
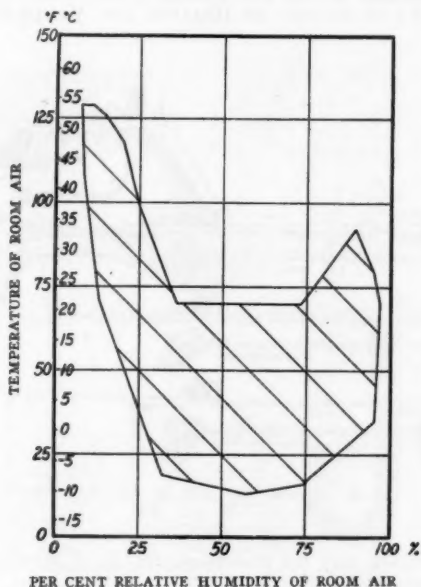
FIG. 2. ROOM AIR CONDITIONS USED IN TESTS

must breathe the air that surrounds them. The physiological consequences of the air inspired must be distinguished from thermal, external effects.

In Part I a brief account of experimental procedures and the results in graphical form will be given. Part II is an indulgence in the human and understandable desire to speculate upon results achieved and further work to be done. It is speculation which induces new undertakings. And presumably it is required from time to time in order to carry on.

PART I

The object of the experiments was to measure the changes in temperature and moisture content of respired air at different positions in the nasal cavity. The experiments required:

(a) an enclosed space with a controlled atmosphere.
(b) a means of securing a sample of respired air.
(c) measurements of said air sample to find its temperature and moisture content.

In the Laboratory of Applied Physiology an enclosed space about 9 ft square and 7 ft high within a larger room was made available (see Fig. 1 for partial view). A 5 hp air conditioning unit was installed for heating or cooling the test room. A mechanically-aspirated psychrometer and an electric oven were placed in the test room. While air conditions were held nearly constant for any one series of tests the various conditions achieved were somewhere within the area shown in Fig. 2. These conditions embrace those used by the American Society of Heating and Ventilating Engineers,
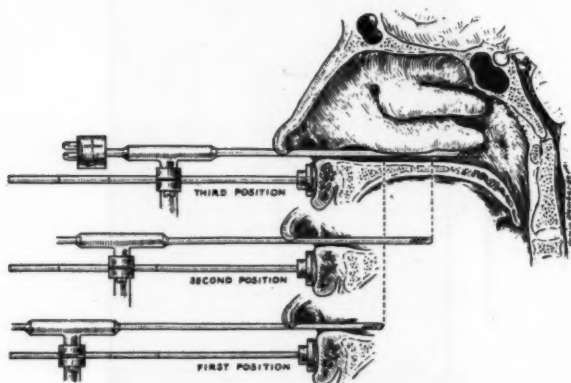
Fig. 3. Sampling Tube in Nasal Cavity

Dr. C.-E. A. Winslow and his associates, Dr. E. F. DuBois and others in their body-heat-loss studies.

The sample of respired air was obtained from the nasal cavity generally between the floor of the cavity and the inferior turbinate by means of a stainless steel tube ⅛ in. in diameter and about 5 in. long. Except for that portion of the tube actually inserted in the nose, the remainder was enclosed in a layer of wool covered with oil silk. The sampling tube was perforated near the end to admit the air sample. Near the perforations, within the tube, was a copper-constantan thermocouple made of very fine wire for quick response. The couple was considered to be adequately shielded from radiation.

The sampling tube was inserted into the nasal cavity in the positions shown in Fig. 3. The positions could be duplicated by means of the adjustable stop shown below the tube. Positions were numbered 1, 2 and 3. The letter I was used to indicate samples taken during inspiration and E for samples during expiration. In addition to the above a position No. 4 (not shown) was used to indicate a test where the tube was out in the portal of the nose for the purpose of taking expired air only.

The sampling tube could be attached or detached from an electrically heated, insulated handle in which a push button valve was located. This valve was manually operated to control the time and duration of taking the sample.

From the handle the air passed through an absorption tube containing an-
hydrous magnesium perchlorate. This substance absorbed the water vapor in
the air sample. The absorption tube was placed in an insulated bag. All
outside parts of the air flow system from the sampling tube to the absorption
tube were either insulated or heated. The sampling tube when not in use
and the absorption tubes were kept warm in an electric oven. The water vapor
was not allowed to condense before the air sample reached the absorption
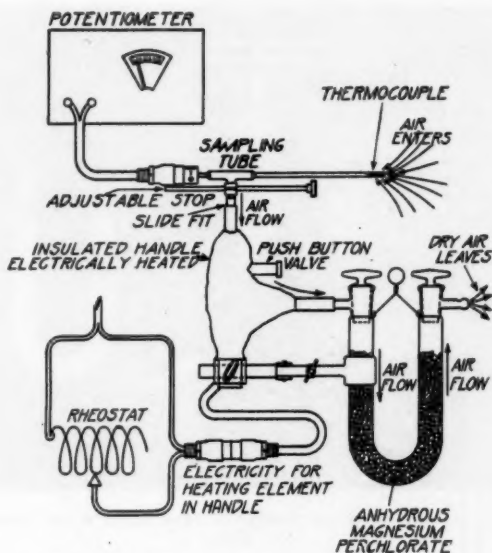
FIG. 4. DIAGRAMMATIC SKETCH OF SAMPLING TUBE
AND ABSORPTION TUBE

tube. The various parts described previously may be seen in Figs. 4 and 4a.
Insulation on the tube is not shown.

After leaving the absorption tube the air, now dry, was conducted to the
larger room to a collecting bottle. The air bubbled up through distilled water
contained therein after passing through some carborundum saturators. These
saturators break the air into tiny bubbles, and an assumption was made
that the air reached the top of the bottle saturated at the water temperature.
Since the bottle was in thermal equilibrium with the air of the outer room, it
was difficult to see how the air sample temperature could be any different from
the bottle and its contents. The volume of air collected had to equal the
volume of water removed from the bottle. Either 1000 or 2000 cc of water
were collected in a graduate. The pressure of the saturated air sample was
measured with a manometer. Knowing the pressure, volume and temperature

of the saturated air, the weight of dry air in the sample could be determined. Figs. 5 and 5a show the assembly described.

From the aspirating psychrometer data the weight of water per pound of room air could be calculated. The absorption tubes were weighed after reaching thermal equilibrium with the air in the room housing the balance. Corrections for change in the buoyancy of the tubes based on barometer and air temperature were made. The difference between the moisture absorbed and the moisture in the original air being breathed gave the moisture evaporated in the
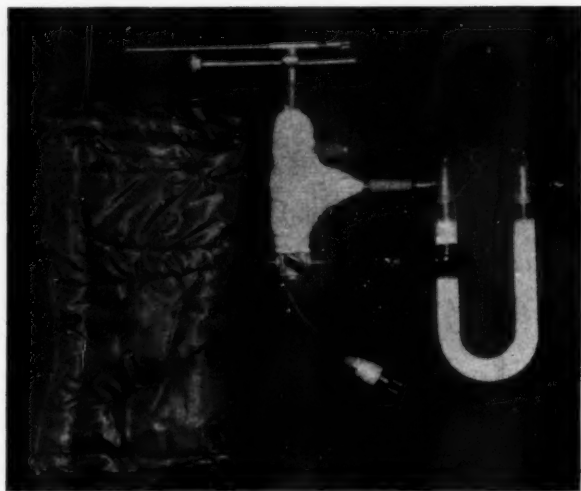
FIG. 4A. PHOTOGRAPH OF SAMPLING TUBE USED WITH ABSORPTION TUBE

nasal cavity. By means of a potentiometer the temperature of the air sample from the nasal cavity was determined. From the given data the relative humidity of the air sample could be calculated.

Results of interest are as follows:

(a) The temperature of the air sample.
(b) The relative humidity of the air sample.
(c) The water evaporated in the nasal cavity per pound of dry air breathed.
(d) The sensible and latent heat of the air sample.

In general, the tests were run in the following manner. Test room conditions were stabilized at the conditions desired. The subject entered the room sometime prior to the test—usually not less than 20 min. The conditions of the nasal cavity were believed to be normal. Tests were run at the three positions indicated in Fig. 3 both for inspired I and expired E air and last of all one test at position No. 4. In *all tests* fairly deep and deliberate breathing was done at a rate which avoided over-ventilation. The subject was standing

and was moderately active.   Tests with shallow breathing will require more convenient temperature measuring apparatus.   The air sample was taken only during the latter part of each inspiration or exhalation in order to get a proper sample.   Apparatus was originally developed to take samples automatically, but manual operation of a valve was found to be satisfactory.

In passing, it should be mentioned that small samples of air cannot be safely conducted through rubber tubing without danger of a change in moisture
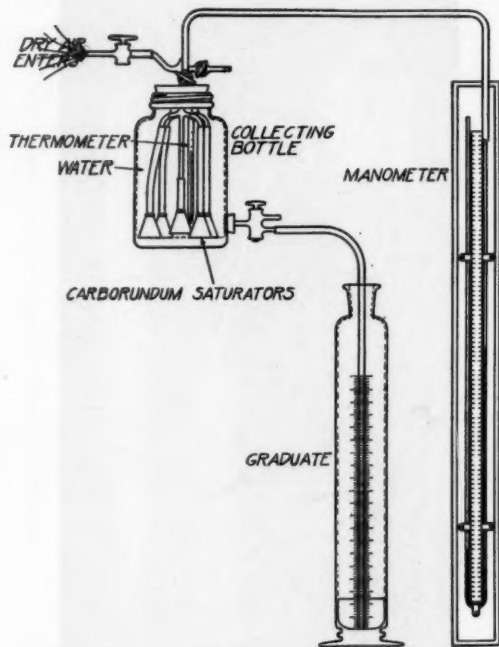
FIG. 5. DIAGRAM OF COLLECTING BOTTLE AND MANOMETER

content.   Rubber was found from certain special tests to be too hygroscopic for accurate results.   Some time was also spent in comparing the psychrometer against the absorption method of determining the moisture content of the air in the test room.   It appears that the psychrometer indicates an amount of water vapor greater than the absorption method gives.   However, if there is any error in psychrometer results it is less than physiological variations.

The air movement in the test room was rather vigorous.   The current of room air reached the psychrometer first and then the subject.   It is important for investigators to keep in mind that they constantly change the room air by involuntarily giving off heat, moisture, carbon dioxide, etc.   Furthermore, it should be noted that some of the common methods of determining relative

humidity will actually change the moisture content of the air used. This would be unimportant unless the total quantity of air used in the tests were small.

Figs. 6 to 8, inclusive, show general results selected from a number of test series. The increase of water vapor may be noted up to position 3*I*. The dashed line indicates a slight increase in water vapor added by the lungs and

FIG. 5A. PHOTOGRAPH OF COLLECTING BOTTLE
AND MANOMETER

interconnecting passageways. The nasal cavity apparently does the major bulk of the air moistening. Nose breathing through a normal nasal cavity allows very little water vapor to escape from the lungs.

From positions 3*E* to 4 it will be noted that some water vapor appears to be recovered or absorbed in the nasal cavity during expiration. Fig. 9 shows the amount absorbed and also the per cent expressed in terms of the amount of water vapor added at position 3*I*. At low temperatures the apparent recovery is probably condensation in part due to cooling but this cannot be true at high room temperatures.

Figs. 6 to 8 also show changes in air temperature. Cold air is heated and

warm air is cooled.   Since breathing is an intermittent process the anterior
of the nasal fossa may be momentarily cooled below or heated above the
average temperature of the surface.   This is shown by the cooling or heating
of the expired air.   Surface conditions apparently change more rapidly than
they can be counteracted during the same interval of time.   When cold air
is breathed there is a recovery of heat from the expired air as well as from
the recovered (condensed) water vapor.

Relative humidity is defined as the ratio of the actual water vapor pressure
(or density) to the saturated water vapor pressure (or density) corresponding
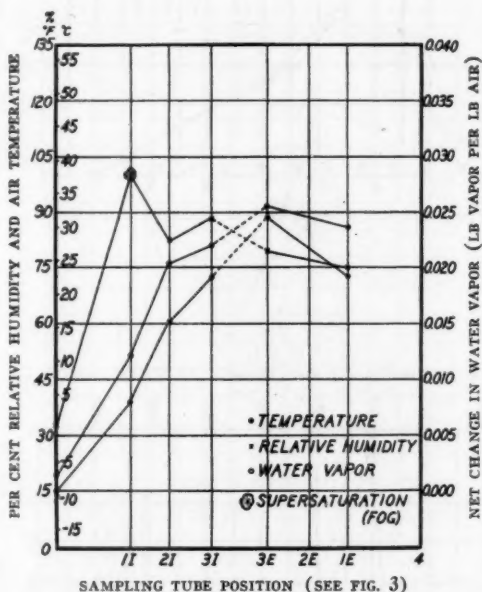
FIG. 6. CHANGES IN TEMPERATURE, RELATIVE HU-
MIDITY, AND WATER VAPOR vs. SAMPLING TUBE
POSITION

to the temperature of the air and its associated vapor (dry-bulb temperature).
It will be noted that air over pure water will become saturated when equi-
librium is reached.   The relative humidity will reach 100 per cent.   However,
air over any solution (i.e., body fluids) in equilibrium never can reach 100
per cent relative humidity.   Saturation, in view of the definition given, can
never occur due to the decreased water vapor pressure of the solution.

The relative humidity curves in Figs. 6 to 8 show the saturated air does
not come from the lungs.   The curves seem erratic but they actually depend
upon the water vapor and temperature readings which look consistent.   It is
possible for the relative humidity to decrease even when the water vapor

increases because of the simultaneous rise in air temperature. Fig. 6 shows that cold air encountering warm, moist surfaces of the nasal cavity may produce a supersaturated or fog-like condition. It is possible in cold air to have a fog zone in some part of the nasal cavity during inspiration and also during expiration.

Fig. 10 shows a more comprehensive picture of temperature changes. Neither at 3*I* nor 3*E* does the air temperature approach a standard value. If the surfaces in the nasal cavity were dry and uniform in temperature, the

TEMPERATURE OF ROOM AIR
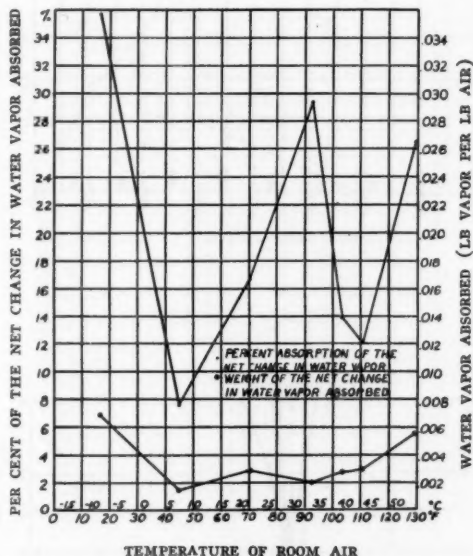
FIG. 9. AMOUNT OF NET CHANGE IN WATER VAPOR AT SAMPLING TUBE POSITION 3*I* (SEE FIG. 3) AB- SORBED BY NASAL CAVITY DURING EXPIRATION *vs.* ROOM AIR TEMPERATURE

final temperature at 3*I* would depend upon the initial room air temperature. This is in accord with the performance of air heating devices. The fact that the nasal surfaces are moist does make it possible for the air temperature to drop and then rise again. A thermometer with a wetted bulb will read lower than a dry-bulb thermometer unless the air is fully saturated.

The temperatures in the various parts of the nasal cavity depend upon the initial room temperature—breathing rates and air velocities being kept reasonably uniform. The invariable rise in temperature between 3*I* and 3*E* indicates that some heat is added by the lungs and interconnecting passageways. The temperatures at position 4 are the ones that should be used to compute the actual heat losses from the respiratory system. An expired air temperature of
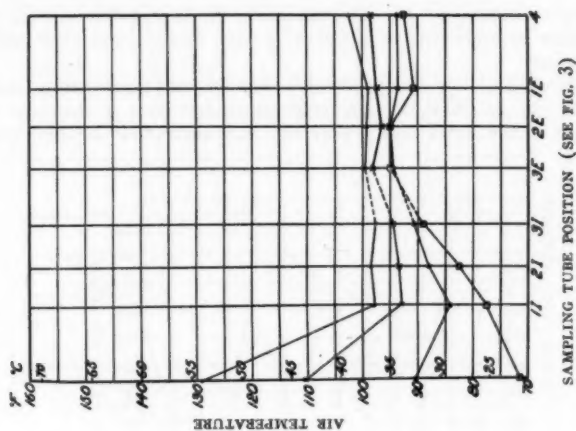
FIG. 11. Air Temperature vs. Sampling Tube Position at Low Relative Humidity (Average 13.4 per cent) of Room Air
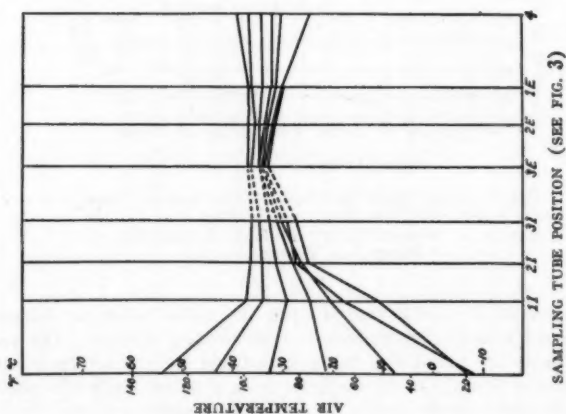
FIG. 10. Air Temperature vs. Sampling Tube Position

about 93 F (33.9 C) would be correct only if the inspired air temperature was about 70 F (21.1 C). The low-temperature curves suggest that very low-temperature air must enter the lungs at a low temperature and take considerable heat therefrom.

Fig. 11 shows temperatures when the room air is low in relative humidity (i.e., 13.4 per cent average). Fig. 12 shows the effect of different relative humidities, especially the *wet-bulb* effect. When air at about 70 F (21 C) is varied in relative humidity the variation of temperatures in the nasal cavity

SAMPLING TUBE POSITION (SEE FIG. 3)
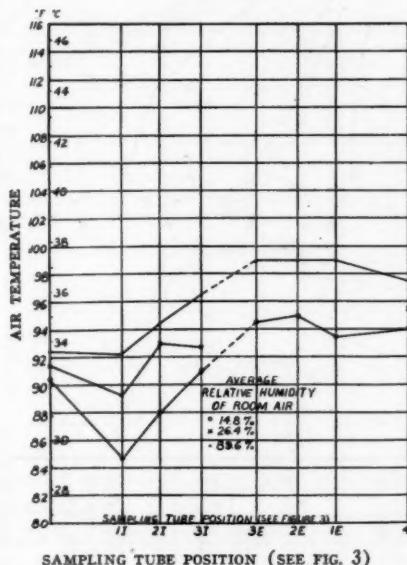
FIG. 12. AIR TEMPERATURE *vs.* SAMPLING TUBE POSITION AT DIFFERENT RELATIVE HUMIDITIES OF ROOM AIR

is not great. Fig. 13 shows that relative humidity has comparatively little effect upon temperatures at position 3*l*. The scale magnifies the differences that do exist though it roughly appears that room air of high relative humidity absorbs heat somewhat faster. This has been found true in tests of air-heating devices.

Taking average air temperatures regardless of relative humidities of room air, Fig. 14 shows temperatures in the nasal cavity at several positions. The area to the left of the dashed line represents air heating and to the right— air cooling. The manner in which the curves cross at high air temperatures reflects the cooling and reheating effects previously mentioned.

Figs. 15 to 17 show the water vapor in pounds per pound of dry air added at the various sampling tube positions at different room air temperatures and
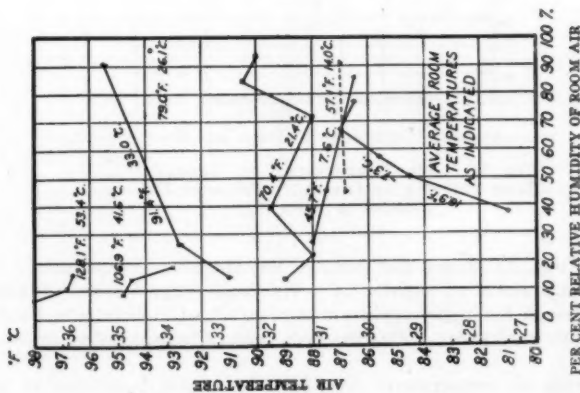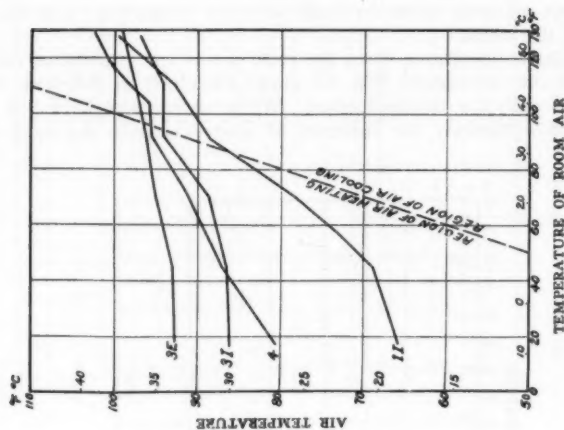
Fig. 14. Average Air Temperature at Sampling Tube Positions Shown vs. Temperature of Room Air

Fig. 13. Air Temperature at Sampling Tube Position 3/ (see Fig. 3) vs. Relative Humidity of Room Air

relative humidities.   Fig. 15 shows most clearly how responsive the warm, moist surfaces in the nasal cavity are to different relative humidities.   Fig. 18 shows the effect of various relative humidities at several average air temperatures for position 3*I*.   The trend in every case is clear.   Fig. 19 shows similar information plotted against the initial water-vapor pressure of the room air. Except for high-temperature, low-humidity air (Fig. 18) the 70.4 F (21.3 C) curve shows rather high evaporation rates.   With one exception the maximum evaporating rate is about 0.025 lb vapor per pound by dry air.   Whether or

SAMPLING TUBE POSITION (SEE FIG. 3)
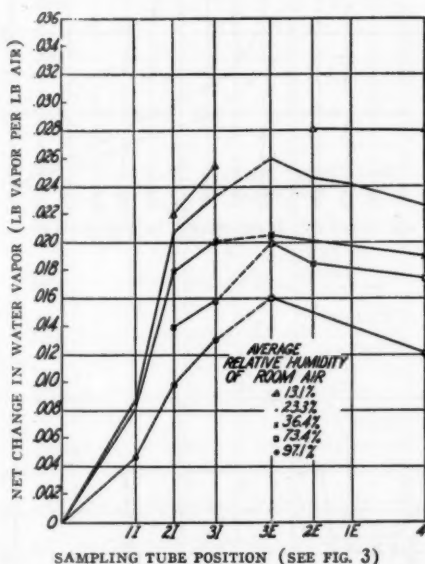
Fig. 15. Net Change in Water Vapor at
Average Room Temperature of 70.5 F (21.4 C)
vs. Sampling Tube Position

not this is a normal maximum cannot be known since the writer has been the only subject experimented upon thus far.

It appears that the most important effect of a sudden change of air conditions may be the equally sudden change in the evaporation rate.   If the nasal cavity continues to be moist, as it should be, the change in evaporation is sudden and unavoidable.

This change in evaporation rate must be compensated for in time in order to maintain a normal mucous lining but how long it takes and what the consequences of even a short misadjustment may be are questions which remain to be answered.   If moisture is evaporated from nasal mucosa at an accelerated rate, due to change of temperature and/or relative humidity, it follows that
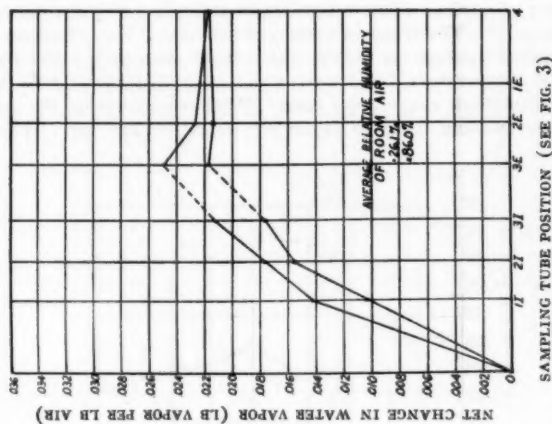
FIG. 17. NET CHANGE IN WATER VAPOR AT
AVERAGE ROOM TEMPERATURE OF 45.6 F
(7.6 C) vs. SAMPLING TUBE POSITION
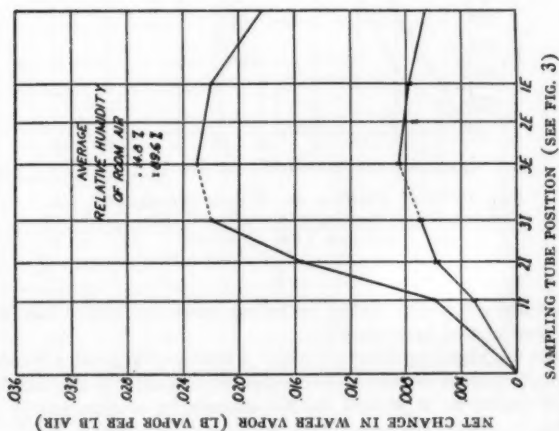
FIG. 16. NET CHANGE IN WATER VAPOR AT
AVERAGE ROOM TEMPERATURE OF 91.5 F (33.1 C)
vs. SAMPLING TUBE POSITION

the characteristics of that mucosa must change in viscosity, surface tension, osmotic pressure, etc., unless a greater supply of water is added to this mucosa or else a greater quantity of mucosa itself is supplied. Unless compensation is instantaneous it must be conceded that at least the physical character of the mucosa is changed by its altered rate of water loss. This may be quite important.

From Fig. 18 it can be seen that going from 18.9 F (-7.3 C) air at 57.5 per cent relative humidity to a 70 F (21.1 C) room at 20 per cent relative
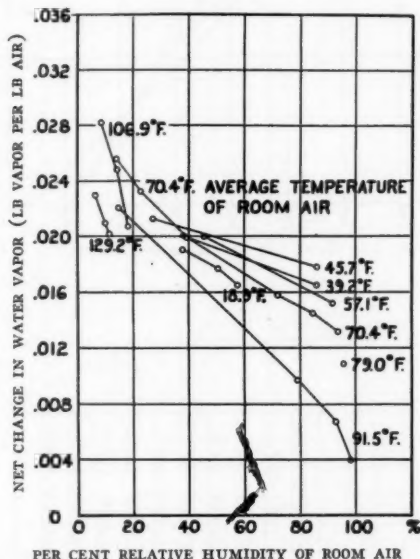
Fig. 18. Net Change in Water Vapor at Sampling Tube Position 3*I* vs. Relative Humidity of Room (see Fig. 3)

humidity changes the evaporation rate from 0.0164 to 0.024 lb vapor per pound of dry air, an increase of 46 per cent. This situation could result when one goes indoors on a cold winter's day. While the physiological effect of a change in evaporation rate may be left to conjecture at the moment, it is true that no climatic or atmospheric changes can equal in speed and severity the change in the above example.

Figs. 20 to 22 show the absolute humidities rather than the net change in water vapor content per pound of air. In general, the greatest change takes place between positions 1*I* and 2*I*. The important fact is that the rate of evaporation per unit of surface is not uniform. A uniform lining of mucosa must require graduations in moisture or mucous supply at different parts of the nasal cavity. Otherwise, certain surfaces would be unduly dry and others
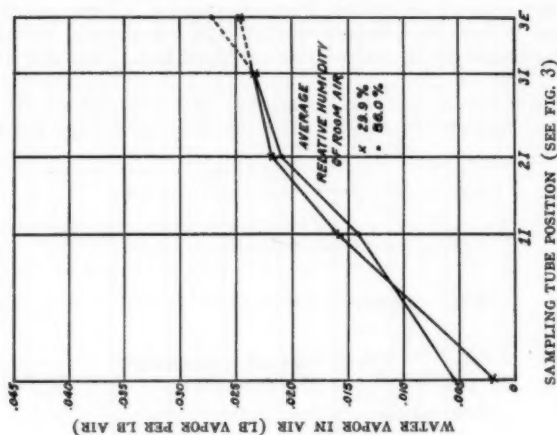
Fig. 20. Water Vapor in Air at Average Room Air Temperature of 45.7 F (7.6 C) vs. Various Sampling Tube Positions
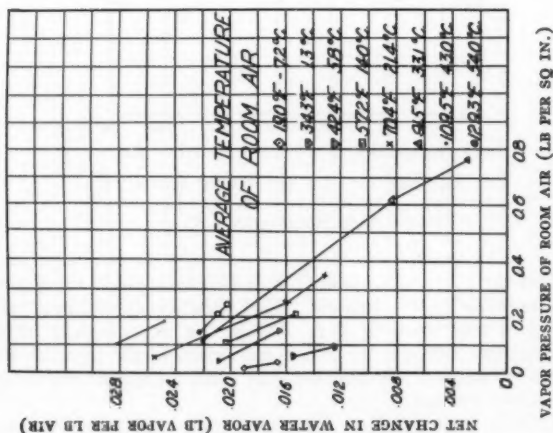
Fig. 19. Net Change in Water Vapor at Sampling Tube Position 31 (see Fig. 3) vs. Vapor Pressure of Room Air

unduly moist. The mucous lining would then be non-uniform. The cause of the differentiation in rate of supply from surfaces that are continuous and similar would be worth knowing.

Figs. 23 to 26 showing changes in relative humidity are of interest principally to show that the air never reaches saturation at the posterior of the nasal cavity (position 3I) nor does it ever appear to be saturated on its return from the lungs (position 3E). The only way to supply saturated air is to have it virtually saturated at body temperature before breathing. (See Fig. 25.)
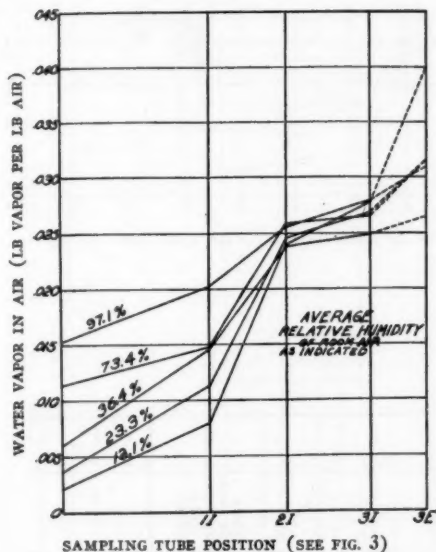
FIG. 21. Water Vapor in Air at Average Room Air Temperature of 70.5 F (21.4 C) vs. Various Sampling Tube Positions

It may be repeated that the air samples were taken at each breath only during the latter part of the period. The air lying in the nasal cavity and passages should have been purged by this procedure before the sample was taken. The fact that higher relative humidities were found at other positions and even supersaturated conditions where they might reasonably occur should lend credibility to the observations even though at positions 3I and 3E they are contrary to general expectations. And again it should be repeated that whenever the water vapor in the air is in equilibrium with body fluids, the relative humidity can never be 100 per cent.

Figs. 27 and 28 show average enthalpy of air plus water vapor at various room temperatures for relative humidities below 50 per cent (Fig. 27) and above 50 per cent (Fig. 28).

FIG. 23. RELATIVE HUMIDITY OF AIR vs. SAM-
PLING TUBE POSITION AT AVERAGE ROOM AIR
TEMPERATURE OF 45.6 F (7.6 C)

FIG. 22. WATER VAPOR IN AIR AT AVERAGE
ROOM AIR TEMPERATURE OF 91.5 F (33.1 C)
vs. VARIOUS SAMPLING TUBE POSITIONS

Fig. 29 shows the enthalpy of the air at various temperatures and relative humidities of air for position 3I. As would be expected, the value changes very little at different relative humidities of the room air. Fig. 30 shows the enthalpy of evaporation and here the values are definitely responsive to variations in relative humidity.

Fig. 31 represents the sum of the values shown in Figs. 29 and 30. It shows the combined effect of the two characteristics previously noted. Fig. 32 gives the total enthalpy of the air plus water vapor at position No. 4. The
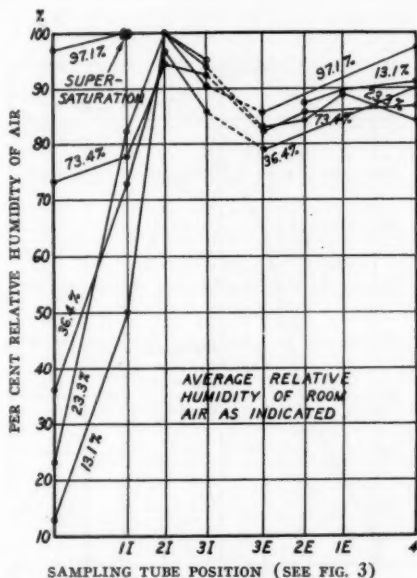
FIG. 24. RELATIVE HUMIDITY OF AIR *vs.* SAMPLING TUBE POSITION AT AVERAGE ROOM AIR TEMPERATURE OF 70.5 F (21.4 C)

curves for average air temperatures of 44.9 F (7.2 C) and 17.6 F (−8.0 C) were made from points which varied more than those values at the higher temperatures shown. It is clear that a determination of external body heat loss must take into account the proper loss due to respiration.

The results presented definitely show that nasal cavity air temperatures depend upon the initial temperature of the air with relative humidity having some modifying effect. There is always some rise in temperature between 3I and 3E even when the initial air temperature is above body temperature. Very low-temperature air undoubtedly throws a heavy air-heating burden on the lungs.

It is clearly shown that an accommodation can be made to wide variations in moisture loss. Also it shows that the response in evaporation depends upon

the initial water vapor pressure of the air even though the air temperature remains constant (Fig. 19). Actually both temperature and relative humidity determine the evaporation loss. Any horizontal line on Fig. 18 will indicate a variety of temperatures and relative humidities that will cause equal or unchanged evaporation rates. The comparatively small amount of water vapor taken from the lungs and the partial recovery of water vapor during expiration are both interesting results.

That the relative humidity of the air at $3E$ is never 100 per cent and often well below 100 per cent will probably be viewed as surprising. Luciani,
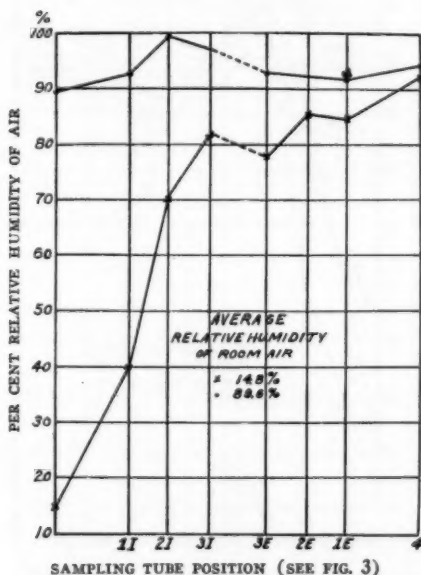
Fig. 25. Relative Humidity of Air *vs.* Sampling Tube Position at Average Room Air Temperature of 91.5 F (33.1 C)

Human Physiology Vol. I, p. 397, states, "The expired air is saturated or nearly so with the aqueous vapour exhaled along the respiratory passages." Also—"Its (expired air) temperature is approximately that of the body (35-36°C)." . . . It will be stated again that the air sample was taken only during the latter part of each inhalation or exhalation and that during this time the air temperatures remained quite steady. Since ventilation is a process of air control by means of dilution there is no good reason why expired air should be saturated. If alveolar air is in equilibrium with the fluid-covered surfaces of the lungs, further evaporation would be inhibited. The only way for water vapor to escape from the lungs would be by diffusion and mechanical mixing with respired air.

Final observations should include comments upon the swelling of the turbinates. In general this was associated with the higher rates of moisture loss from the nasal cavity. Also an occasional or momentary swelling might occur and pass during a test. It is the opinion of the writer that this may have been the local response to a momentary vaso-dilation due to nervousness or fatigue. If a general vaso-dilation or constriction has some corresponding effect upon the turbinates, an interesting speculation may be made. During the momentary condition above mentioned the variations in temperature from

SAMPLING TUBE POSITION (SEE FIG. 3)
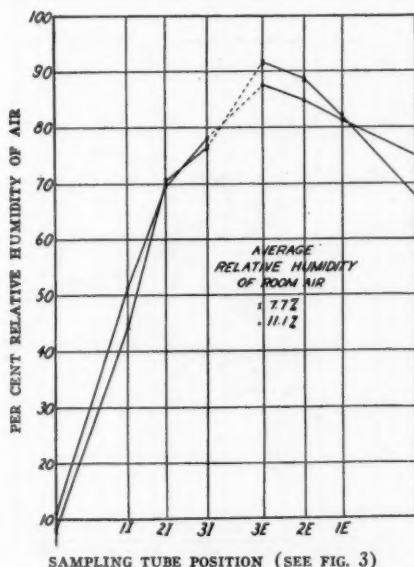
FIG. 26. RELATIVE HUMIDITY OF AIR vs. SAMPLING TUBE POSITION AT AVERAGE ROOM AIR TEMPERATURE OF 129.3 F (54.0 C)

breath to breath were appreciable. Without such disturbances the temperatures were remarkably uniform.

At least one must concede that the nose is a very effective air conditioning device.

PART II

The results given in Part I induce a desire to explain or speculate upon them. In the first place, the recovery of water vapor during expiration ought to be explainable on the ground of differences in water vapor pressure. It must be constantly realized that the nasal mucosa is being intermittently robbed of its water. An inhalation removes moisture with a speed which probably exceeds the ability of the body to counteract. If the water content of mucosa

is temporarily lowered, the partial pressure of the water in the mucosa would also be lowered. The expired air does acquire some water vapor from the lungs and therefore its water vapor pressure will be increased. In such a case any tendency towards equilibrium would require a drop in the water vapor pressure of the expired air. This would involve the condensation of some of the water vapor in the air upon the mucous surface. Peters, in "Body Water" states "In theory, at least, then, the amounts of water lost by vaporization should be influenced by the osmotic pressure of tissue fluids at the exposed surfaces of the body." The possibility of condensation upon the
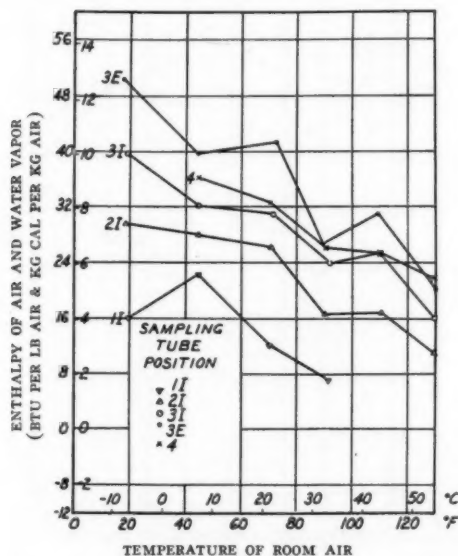
Fig. 27. Average Total Enthalpy of Air and Water Vapor at Sampling Tube Positions Shown (see Fig. 3) vs. Temperature of Room Air

tissue fluids must also be conceded in theory, at least. At any rate, the tests do show a recovery of water vapor during expiration.

It does seem curious that temperatures at 3E should exceed those at 3I when the initial air temperatures are high (i.e., 130 F—54.4 C). While the temperature rise is slight it does suggest that small quantities of energy may be involved in the gaseous exchanges effected in the lungs and the physio-chemical changes associated therewith or that it is merely some of the heat given off by oxidation in live tissues.

The statement is commonly made to the effect that the normal nasal cavity is covered with a continuous film of mucosa, strong enough to be drawn slowly

back to the throat by means of ciliated epithelium. Since mucosa must suffer a water depletion in its progress, the question arises whether the mucosa arrives at the back of the nasal cavity in an altered condition or not. It must be evident that a loss of moisture will affect viscosity, surface tension and osmotic pressure. If the loss of moisture is high it is conceivable that the mucosa would become too viscous to be moved by the cilia. This would destroy the cilia, produce discomfort and uncover tissues beneath the cilia to infection. If surface tension has any bearing upon the recognized germicidal characteristics of nasal mucosa then an excessive loss of water might deprive mucosa
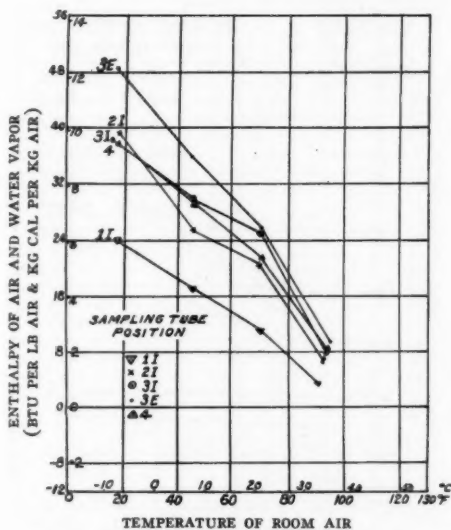
FIG. 28. AVERAGE TOTAL ENTHALPY OF AIR AND WATER VAPOR AT SAMPLING TUBE POSITIONS SHOWN (SEE FIG. 3) vs. TEMPERATURE OF ROOM AIR

of its protective qualities. The change in osmotic pressure could induce a greater flow of liquid through membranes adjacent to the mucosa if osmotic interchange of fluids applies to the nasal cavity. This principle, if it is involved, would tend to correct or compensate for water loss.

It is reasonable to suppose that the nasal mucosa would retain certain normal characteristics all over the nasal cavity. In this case, however, it would be necessary in some manner to replenish the mucosa with the water that it is constantly losing. Merely adding mucosa along the path of motion would not maintain uniformity of viscosity, etc., though it would prevent absolute drying. However it is done, it is clear that supposedly normal

mucous membranes can be preserved over wide differences in moisture loss which depend upon the temperature and relative humidity of the inspired air.

However, the most interesting consideration is the effect of a sudden change in air conditions. Figs. 18 and 19 show how the moisture loss would rise upon entering a dry, warm room from the cold outdoors. This suddenly altered rate of evaporation may change the viscosity, osmotic pressure and surface tension quickly with whatever physiological consequences may flow therefrom. Destruction of cilia or loss of germicidal protection in the presence of infectious organisms may result before the body accommodates
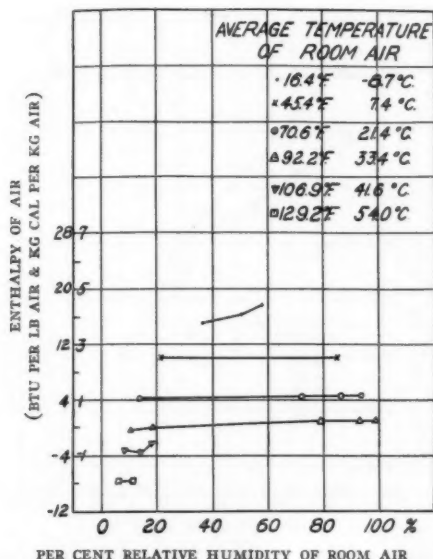
FIG. 29. Enthalpy of Air at Sampling Tube Position 3I (see Fig. 3) vs. Relative Humidity of Room Air

itself to the new requirements. In such a case one could *catch* a cold upon entering a warm room where infection is present. It does not seem likely that the body could compensate for the accelerated water loss to avoid a brief but definite change in the physical characteristics of the nasal mucosa. It is proposed to ascertain the time required for this adjustment in future work.

It is probable that the excess of nasal mucosa experienced when passing from a warm, dry room to the cold outdoors is not due to increased production but rather to a lower rate of evaporation. The adjustment to this condition takes a number of minutes. The watering of the eyes might be caused in the same way since the surface of the eyes must be kept moist at all times.

If this sudden alteration in moisture loss from the nasal cavity is undesirable it follows that air conditioning ought to be governed to avoid drastic changes. Fig. 18 shows that indoor spaces should be humidified in cold weather. Horizontal lines in Fig. 18 would indicate what should be done. For moderately cold weather, relative humidities of indoor air at 70 F (21.1 C) ought to vary between 40 to 60 per cent. This is what everybody knows already. This effort merely provides a reason—a good one, it is hoped.

However, if one considers summer cooling of buildings one must conclude

PER CENT RELATIVE HUMIDITY OF ROOM AIR
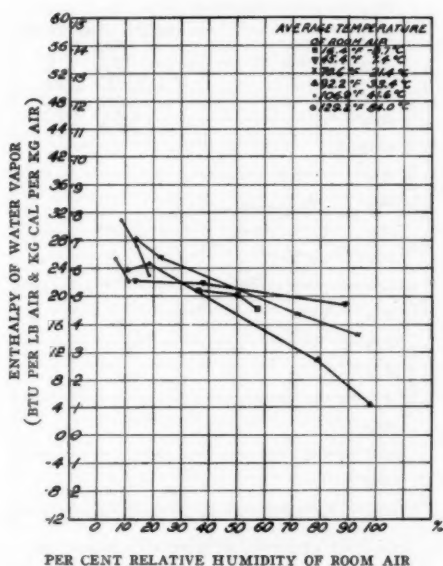
FIG. 30. ENTHALPY OF WATER VAPOR AT SAMPLING TUBE POSITION 3*l* (SEE FIG. 3) *vs.* RELATIVE HUMIDITY OF ROOM AIR

that the nature of the climate should decide the character of cooling. If the climate is hot and dry, the method of cooling should provide a dry indoor atmosphere. If warm and humid then rather moist indoor air would be better. These observations are based on the premise that drastic changes in moisture loss from the nasal cavity must be avoided where the change in environment is sudden.

If nasal mucosa serves mainly to protect the respiratory system against dust and infection it then appears that the moistening of the air is an unavoidable accompaniment. It is certain that the lungs function with tidal air below 100 per cent relative humidity and at no standard or fixed value.

It is believed that the humidification of the inspired air may be more for protecting the throat from irritation than for any other reason. The consequences of mouth breathing, speaking and singing support the belief. It would be interesting to know if bronchitis may not be induced or at least aggravated by some failure of the mucous membrane to adequately humidify the inspired air.

In considering the elimination of carbon dioxide from the lungs Grandis (1900)[1] pointed out that water vapor is eliminated by expired air. He

AVERAGE TEMPERATURE
OF ROOM AIR
. 16.4°F -8.7°C
× 45.4°F 7.4°C
: 70.6°F 21.4°C
▽ 92.2°F 33.4°C
▲ 106.9°F 41.6°C
■ 129.2°F 54.0°C

ENTHALPY OF AIR AND WATER VAPOR
(BTU PER LB AIR & KG CAL PER KG AIR)

PER CENT RELATIVE HUMIDITY OF ROOM AIR
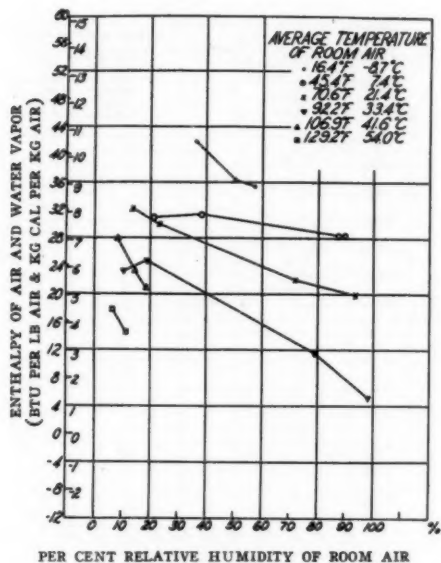
Fig. 31. Enthalpy of Air and Water Vapor
at Sampling Tube Position 3*l* (see Fig. 3)
vs. Relative Humidity of Room Air

assumed that this water vapor originated in the lungs and pointed out that the blood must undergo a temporary increase in concentration during its passage through the lungs which raises the carbon dioxide tension and facilitates expulsion of CO_2 to the alveolar air. This increase in tension (pressure) was verified experimentally. What Grandis pointed out could not apply in the lungs but it could in the nasal cavity.

The blood in the nasal cavity must in some manner make up the moisture loss from the nasal mucosa. When this loss is large there must be an appreciable change in blood concentration and carbon dioxide pressure. Whether the swelling of the turbinates and the general discomfort induced

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[1] Luciani, Vol. I, p. 391.

is related to blood concentration, carbon dioxide pressure or the mechanical problems of a fluid exchange or is solely the result of a general vasodilation is a question which might be answered by suitable observations. It was noted in Part I that the swelling of the turbinates was generally associated with high moisture loss in the nasal cavity. It should be kept in mind that all of the tests were run while the conditions in the nasal cavity were assumed to be in equilibrium with the respired air.

With one exception, the maximum moisture loss was about 0.025 lb of
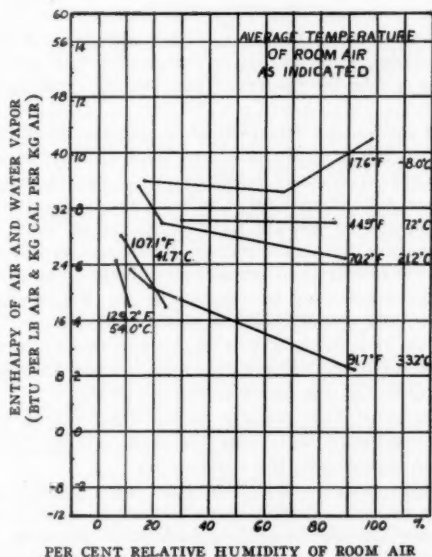
FIG. 32. ENTHALPY OF AIR AND WATER VAPOR AT SAMPLING TUBE POSITION 4 (SEE FIG. 3) vs. RELATIVE HUMIDITY OF ROOM AIR

water vapor per pound of air. This may be a limit characteristic of the person who was tested. It is reasonable to suppose that others might have different limits and that some might be definitely handicapped from a moisture-loss standpoint when the air conditions are severe.

If a vaso-constriction due to sudden chilling has a corresponding effect in the turbinates it would be interesting to discover if equilibrium in moisture loss is disturbed. A temporary deficiency in the supply of mucosa or of moisture supply thereto would induce a temporary change in the physical characteristics of the mucosa in the same manner that a sudden change in air conditions might do. The consequences, whatever they may be, can follow in the same way. If a change in the protective qualities of mucosa or the

completeness of the mucous film may allow infection then one might "catch a cold" due to sudden chilling.

The act of sneezing suggests itself as an attempt perhaps by a sudden mechanical pressure in cells and glands to increase the quantity of mucosa quickly. If mucosa is a defensive agency the reason for sneezing is clear. Some mechanical effect (irritation) due to quick shrinking of the turbinates, breaking or spotty drying of the mucous film, changes in viscosity, etc., might be enough to induce sneezing. The quick increase in the supply of mucosa supports the opinion that, without sneezing, a longer period of time would be required to establish a new equilibrium due to the changed conditions.

In an article on influenza,[2] the following statement is of interest: "One catches influenza *only* through the nose, throat, windpipe or lungs. It is the mucous membrane lining these organs of respiration that provides the portal of entry." Could it not be the failure of the mucous membrane which permits infection?

In discussing an address by Dr. Dochez before the members of the International Congress of Microbiology in New York (1939) on the wide differences in the virulence of diseases affecting the upper respiratory tract, Mr. Gray says, ". . . we may imagine that certain variations in the environment of these invisible agents of disease have an effect on their potency. It is conceivable, for example, that slight changes in temperature, in blood alkalinity, in the contents of the cellular and intracellular fluids or in other bio-chemical factors affecting the virus, may be responsible for the strange shifts which produce new strains, and, it may be, new species of virus." The experiments reported in Part I do at least show that changes in temperature can occur and that changes in the moisture content of nasal mucosa can also occur for a time when the condition of the air is suddenly altered.

It should be mentioned that changes in moisture loss in the nasal cavity due to altered air conditions will exert the same relative effects upon fluids discharged from the sinuses into the nasal cavity.

It is of interest to mention that 300 years ago the use of a *tempering chamber* was recommended. Persons were advised to spend from 5 to 10 min in a chamber having air conditions intermediate in character. An unheated hallway in a heated house was suggested as an example of a *tempering chamber*. This recommendation is not suited to the tempo of modern times, however sensible it might be. Today persons are subjected to what amounts to many different climates in a most abrupt manner. Acclimatization takes time.

Future tests will be made to see if the response in the nasal cavity to different evaporation rates depends solely upon the condition of the inspired air or not. Also an attempt will be made to discover how long it takes to establish a new equilibrium associated with suddenly altered air conditions. Tests employing shallow breathing, the effect of nasal preparations, smoking, etc., suggest themselves. Also tests on persons suffering from apparently upper respiratory or even throat ailments might be deemed of interest. The primary object was to discover or establish more definitely the importance of relative humidity in the field of air conditioning.

While the test results were associated rightly or wrongly with certain physio-

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[2] The Problem of Influenza, by George W. Gray. (*Harper's*, January, 1940.)

logical responses it is at the same time realized that these responses might be produced by other means also. The testing of air conditioning devices is an engineering job. The tests herein were strictly performance tests of an air conditioning device (viz., the nose). The speculations are perhaps in the field ". . . where angels fear to tread."

In conclusion, no one today denies the importance of maintaining a complete and normal mucous lining in the respiratory tract. Anything that, even for a short time, can change the normal qualities of mucosa or the uniformity of its distribution must merit considerable study.

ACKNOWLEDGMENT

DISCUSSION

W. L. FLEISHER: Professor Seeley's paper presents, from a new angle, the whole subject of the requirements of relative humidity and the pathological reactions of human beings to their environment.

For many years the controversy has raged over the necessity of atmospheric moisture in the ability of humans to adjust themselves comfortably or safely to winter conditions. Although the prevalence of colds is much more noticeable in the winter than in the summer, the reasons are obscured by the inability of the scientists to locate the basic causes.

With very little assistance from outside sources, Professor Seeley, using himself as a guinea pig, exerting the utmost patience and bringing a profound philosophy to the subject, has discovered that the saturation of the air taken into the lungs is accomplished by the absorption of moisture primarily from the nasal passages, and not from moisture extracted from the lungs. He has discovered that if the call is great (due to very dry air breathed into the nasal passages), that the mucous membranes become congested and are easily infected; that rapid changes of environment, particularly when the mucous membrane is in a congested state, decreases the resistance of these parts and allows bacteria and virus to lodge in the membranes and grow into large colonies; that there are limits of safety that can and should be maintained.

In my opinion he has certainly set up a thesis for the requirements of a broad band of relative humidities, that cannot be overlooked or go unheeded. No more important beginning of a profound idea pertaining to the health of the human race has been presented to the Society. I hope that not only Professor Seeley, but others will carry forward this splendid investigation that he has been carrying on so successfully.

JOHN JAMES (WRITTEN): The maintenance of proper relative humidities in residences or other enclosures continues to be a controversial question. The practical upper limit for interior relative humidity is now established on the basis of current building construction where the condensation of moisture on the interior surfaces is the

controlling criteria. Obviously, structures with insulated walls and double glazing can maintain higher relative humidities than more poorly constructed buildings.

The beneficial effects of humidity on the health and well being of adults have not yet been proved. Dry air is known to have a peculiar sensation on the mucous membranes, but whether the resultant effect is detrimental to health, no one knows. The author seems to have approached this problem logically by studying effects on nasal cavity as relatively dry and wet air is inhaled and exhaled. It is hoped that this work can continue and that other investigators will be inspired to study this important problem from different angles.

REACTIONS OF 745 CLERKS TO SUMMER AIR CONDITIONING

By W. J. McCONNELL,* B.S., M.D., AND M. SPIEGELMAN,** M.E., M.B.A.
New York, N. Y.

PURPOSE OF STUDY

THIS study was undertaken primarily for the purpose of ascertaining the cooling requirements for summer air conditioning in the new unit of the home office buildings of the Metropolitan Life Insurance Co. It was suggested by the Research Technical Advisory Committee on Sensations of Comfort of the American Society of Heating and Ventilating Engineers, whose interest was essentially to obtain *conclusive information on the indoor requirements for different geographical regions of the country.* The study was conducted during the summer of 1938.

The first studies on the cooling requirements for air conditioning were made by the Society in its Research Laboratory located in Pittsburgh, Pa. Later, other studies were made in cooperation with the Society in Texas and Ontario, with a limited number of college students as subjects.[1,2] In order to learn whether the reactions of the college students were typical of persons of both sexes in the occupational ages as they went about their daily tasks, another study was made in an air conditioned office in Minneapolis with 275 employees.[3]

The present study covers 745 clerks (172 men, 573 women) on seven floors of the new unit of the home office of the Metropolitan Life Insurance Co. Although the ages of the clerks were not ascertained, a variation between 17 and 65 years is possible.

Method of Collecting Data

The clerks on the floors selected for study were asked in a circular, shown in Fig. 1, to cooperate on a voluntary basis. In this circular, a description was given of the purpose of the study and the means by which it would be carried out. The perfect cooperation which was secured eliminated at least one source

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Air Conditioning Study

The Company is making a study to determine the conditions of temperature and humidity in the office under which our people are most comfortable. The most important requirement of the study is the collection of facts showing the physical reactions of a large number of people from day to day. Your Section is one in which the study will be conducted. We are, therefore, asking your assistance to the extent of filling in daily the information called for on a form card, a sample of which is attached. The Company believes that everyone will wish to cooperate in the study, but it should be distinctly understood that participation in it is entirely voluntary.

No special instructions for filling in the cards are necessary. Attention is called, however, to the last item where space is provided for general comments which will help in interpreting the record. Here you are asked to make note of any special conditions which may account for a feeling of discomfort in the air-conditioned offices, as for example, "severe warmth at 3 P.M. due to sun's rays"; "not feeling up to par"; "more than usual physical activity due to walking to and from files around 11 A.M."; "sitting too close to walls"; "feel a draught"; etc. Note should also be made here of a distinct feeling of discomfort at any other time of day, giving the time and a statement as to possible causes. If during the period of the study your location in the Section is changed, a record of that fact should be made in the space for general remarks.

Cards will be distributed each morning and collected the following morning.

Your recorded reactions will enable us to tabulate and analyze the group feelings. We cannot stress too much that a record of your personal feelings of warmth or chilliness, uninfluenced by comparison with others, and wholly free from personal bias, is what is needed to make the work of greatest value.

Your cooperation in this work will be a distinct contribution to the knowledge necessary for more satisfactory air conditioning.

Personnel Officer.

Fig. 1. Reproduction of Circular Describing Study

of bias, namely, that which may arise from the inclusion only of those persons careful of their health.

The period of observation extended from July 6 to September 30, 1938. Every morning each clerk received a questionnaire card which was collected

the following day. For purposes of statistical control, section heads were asked to return the cards daily for clerks absent. A copy of the card used is shown in Fig. 2. In addition to name, section, date, and sex, the clerk was asked to place a check in the appropriate column describing his or her reaction at a stated time of day. In the relatively few cases where a clerk made out the card incompletely, she was asked to recall her reaction to the best of her ability. Although the card asked for the reaction of the clerk to conditions outside of the working hours, tabulations were made only on the reactions during working hours, namely, 11 a.m., 3 p.m. and 4 p.m. Special comments noted at the bottom of the cards were studied daily by the engineers in charge of the air conditioning equipment.

RECORDS OF PHYSICAL CONDITIONS

At first, the clerks were asked to designate on the card, by prescribed symbol, the area of the floor in which they were located. This step was

RECORD OF PHYSICAL REACTIONS TO ATMOSPHERIC CONDITIONS
July, August, September 1938
METROPOLITAN LIFE INSURANCE COMPANY

	Ideal	Chilliness		Warmth	
		Mild	Severe	Mild	Severe
1. Morning reaction to outdoor air on leaving home					
2. Reaction upon entering Section					
3. Reaction during working hours: 11 a.m.					
3 p.m.					
4 p.m.					
4. Reaction to outdoor air immediately after leaving building					
5. Continued reaction during evening					

Male ☐ Female ☐
PLEASE CHECK THUS √ IN APPROPRIATE COLUMN

6. General remarks

FIG. 2. QUESTIONNAIRE CARD

discarded when the records of atmospheric conditions were found to show no appreciable variation in the different areas into which the floor was divided.

Continuous recording thermometers were installed to record the dry- and wet-bulb temperatures during the period in which the tests were conducted. In addition, wet-bulb and dry-bulb temperature readings were taken on each floor by means of a sling psychrometer every day at 11 a.m. and 3 p.m. For the locations in which the readings were taken on a typical floor, see Fig. 3. Outdoor readings were taken daily at 9 a.m., 11 a.m., and 3 p.m. A daily record was also kept of the maximum and minimum outdoor temperatures, together with the times at which they were noted.

In order to obtain more complete reaction data, the air conditions were varied to an appreciable extent both below and above the conditions usually prevailing. This, of course, had the effect of bringing more forcibly to the attention of the clerks the presence of uncomfortable conditions.

Method of Analysis

The daily cards from the divisions on each of the seven floors were sorted separately; first, according to sex, and then, according to reaction (ideal, mild chill, severe chill, mild warmth, and severe warmth) at 11 a.m., at 3 p.m. and at 4 p.m.   For each division, the number reporting daily in each sex, reaction, and time category was then entered in a table on a line corresponding to the average of the effective temperatures observed in the various areas of the division at the time specified.

Effective temperature (ET) is an arbitrary index of the relative feeling of warmth or cold experienced by persons in response to temperature, humidity, and air motion, as developed and used by the ASHVE.

The upper panel of Table 1 shows, by way of example, the results of a tabulation of the cards of women according to their reactions at 11 a.m. on August 10.   In the lower panel of the table, the numbers of each line corresponding to a specified effective temperature have been expressed as percentages of the total number on that line.   That effective temperature for which the per cent ideal is highest is regarded as the optimum effective temperature for the day.   In the case of this example, the highest per cent reporting ideal is found opposite effective temperatures in the range from 70.2 to 70.6 deg.

The daily records were consolidated into monthly records and finally into a summary for the whole period of observations.

Description of Results

Daily Record

For purposes of study, it was convenient to represent graphically the following data for each day during the period of observation (see Fig. 4):

1. Indoor effective temperatures
   a. optimum
   b. maximum
   c. minimum

2. Outdoor effective temperatures
   a. 9 a.m.
   b. mean of 9 a.m., 11 a.m. and 3 p.m.
   c. maximum
   d. minimum

It should be noted, first of all, that during the greater part of the period of observation the range between the maximum and minimum indoor effective temperatures was very limited.   In 29 out of the 62 days of observation, the range was less than 3 F; it was greater than 5 F for only 6 days out of the total.   For 27 out of the 62 days, the range from minimum to maximum was from 3 to 5 F.

As regards the daily range of indoor effective temperature, the period from July 20 to September 28 may be divided into two sub-periods.   In the first, extending from July 20 to August 22, there were only two days when the daily range in the indoor effective temperature was as much as 3 deg.   Thus, it is obvious that during this period there was little room for variation in reaction to changes in air conditions.   During the second sub-period, from

August 23 to September 28, except for two days, the daily range from minimum to maximum indoor effective temperature was over 3 deg. In order to bring out more clearly the foregoing observations, the maximum and minimum daily indoor effective temperatures, as well as the daily optimum effective temperature, have been plotted in Fig. 4.

From a cursory inspection of Fig. 4, it is difficult to observe any obvious correspondence between optimum indoor effective temperature and outdoor effective temperature, whether at 9 a.m., mean of 9 a.m., 11 a.m. and 3 p.m.,

Ⓢ Indicates position of Sling Psychrometer

Ⓡ Indicates position of Recorder
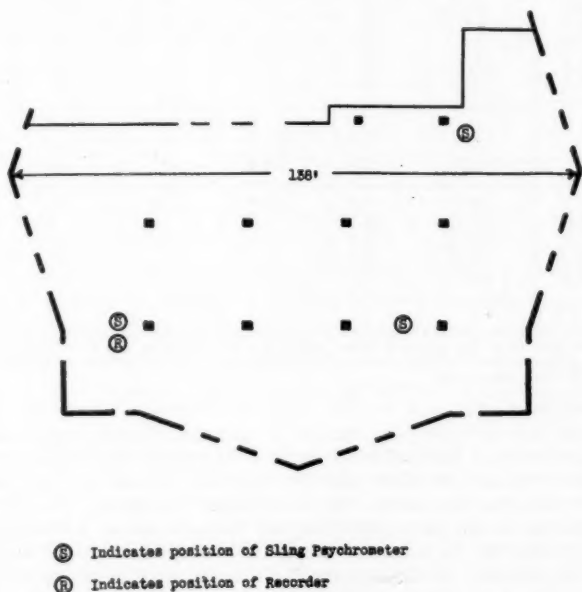
Fig. 3. Plan of One of the Floors Used in the Study

maximum, or minimum. However, an examination of the day-to-day changes, of which a total of 49 are possible,[4] shows that in only 13 instances were the changes in optimum effective temperature different in sign from the changes in the mean of the 9 a.m., 11 a.m., and 3 p.m. outdoor effective temperature. In only 24 out of the 49 possibilities were increases in mean outdoor temperature accompanied by increases in optimum indoor temperature, or decreases in outdoor temperature accompanied by decreases in optimum temperature. For the remaining 12 out of the 49 possibilities, there were no day-to-day changes in optimum indoor effective temperature.

[4] The five-day working week and intervening week-ends reduce the number of day-to-day changes possible to 49, although there was a total of 62 days of observation.

TABLE 1.   TABULATION OF REACTIONS AUGUST 10, 1938, 11 A.M.

EFFECTIVE TEMPERATURE DEG. FAHR.	WOMEN					
	All Conditions	Ideal	Chilliness		Warmth	
			Mild	Severe	Mild	Severe
	NUMBER					
Total	472	313	18	3	127	11
69.8–70.2	83	63	9	3	8	...
70.2–70.6	18	16	1	...	1	...
70.6–71.0	62	45	1	...	16	...
71.0–71.4	227	146	7	...	66	8
71.4–71.8	82	43	...	...	36	3
	PER CENT					
Total	100.0	66.3	3.8	0.7	26.9	2.3
69.8–70.2	100.0	75.9	10.9	3.6	9.6	...
70.2–70.6	100.0	88.8	5.6	...	5.6	...
70.6–71.0	100.0	72.6	1.6	...	25.8	...
71.0–71.4	100.0	64.3	3.1	...	29.1	3.5
71.4–71.8	100.0	52.4	...	...	43.9	3.7

This table accounts for only 472 women out of a total of 573 present when the study was started.   The difference between these two numbers arises principally from absences and from clerks on vacations. Some of the difference may also arise on account of clerks who left or who entered the employ of the company since the study was started.

A comparison of day-to-day changes in optimum effective temperature with the changes in 9 a.m. outdoor temperature is of special interest, for it indicates the extent to which the latter may be used as a gauge by which to adjust the air conditioning equipment daily for optimum conditions.   Here it is found that in 14 out of the 49 possibilities, the optimum indoor temperature made a change from the previous day which was contrary to that made by the outdoor temperature; in 23 instances the changes were in the same direction, and in the remaining 12, the optimum temperature did not change from that of the previous day.

RELATION BETWEEN DAILY OUTDOOR TEMPERATURE AND INDOOR OPTIMUM EFFECTIVE TEMPERATURE

For purposes of investigating further the possibility of a relationship between daily outdoor temperatures and indoor optimum effective temperature, reference should be made to Figs. 5, 6, 7 and 8.   In none of these figures will there be observed any definite correlation between optimum indoor effective temperature and outdoor dry-bulb temperatures, whether they be at 9 a.m., the mean of 9 a.m., 11 a.m., and 3 p.m., the maximum, or the minimum.   This negative finding is not particularly surprising in view of the narrow range of the daily indoor effective temperatures.   This limitation in the present study
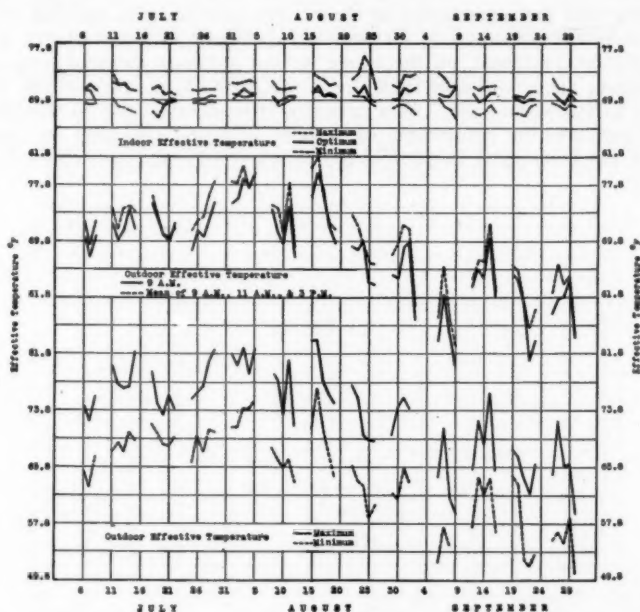
FIG. 4. DAILY TEMPERATURE LOG AND OPTIMUM EFFECTIVE TEMPERATURE FOR INDOOR AIR CONDITIONING JULY, AUGUST AND SEPTEMBER, 1938

made it impossible to confirm the conclusion stated in a previous report,[5] namely, *that a relationship exists between the optimum effective temperature and the four daily weather conditions against which they are plotted*. The four daily weather conditions referred to in this quotation are outdoor maximum, minimum, and mean dry-bulb temperatures and the average of the outdoor mean dry-bulb temperature for a given day and the two previous days.

REACTION CURVES FOR ENTIRE PERIOD OF OBSERVATION

Although the daily range in indoor effective temperatures was limited, the range over the entire period of observation was rather wide, covering the interval from 67.5 to 75.6 deg.

When the daily reports for the entire period were consolidated, the results shown in Fig. 9 are obtained. The curves in this figure are typical of the reactions of comfort, warmth and chilliness to variation in effective temperature. As is to be expected, with advance in effective temperature, the curve for *chilliness* falls off rapidly while that for *warmth* mounts sharply. The curve for comfort is hat-shaped and rises to a maximum in the interval 69.8 to 70.1 deg mean effective temperature, at which 66.2 per cent of the persons

[5] Loc. Cit. See Note 3.

reporting over the entire period of observation noted an *ideal* reaction. There-fore, under the conditions of the present experiment, the interval 69.8 to 70.1 deg may be considered, on the whole, as the optimum effective temperature for summer air conditioning in the new unit of the home office of the Metro-politan Life Insurance Co. As a curiosity, it is observed that in this interval of temperature, the percentages of persons reporting chilliness and warmth are not far from equal, being 15.3 per cent and 18.5 per cent, respectively.
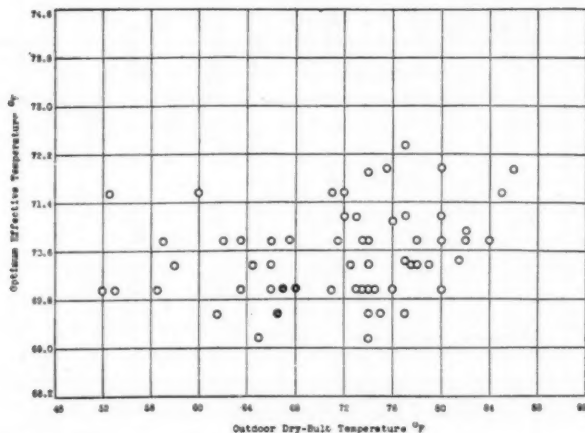
Fig. 5. Relation between Daily Outdoor Dry-Bulb Tempera-ture at 9 a.m. and Optimum Effective Temperature July, August and September, 1938

Reactions of Men and Women Compared

As observed in a previous study,[6] it is found that men prefer a slightly lower indoor temperature than women. According to the comfort curves for the entire period of observation shown in Fig. 10, the greatest proportion of comfort votes from men, 77.8 per cent, is found in the interval 68.6 to 68.9 deg ET. For women, a maximum of 65.0 per cent comfort votes is recorded in the interval 69.8 to 70.1 deg ET. The difference in optimum effective temperature between men and women, in this case 1.2 deg, is pre-sumably related to differences in clothing habits.

There is one further observation to be made under this heading. It will be seen in Fig. 10 that the curve for men falls below that for women at only three points, and then just by small margins. Although it has been stated in the preceding paragraph that men prefer a somewhat lower temperature than women, the present observation suggests that men are not as susceptible as women to discomfort, especially at the extremes of temperatures recorded in this study.

[6] Loc. Cit. See Note 3.

REACTIONS IN MORNING AND AFTERNOON COMPARED

For practically the entire range of temperatures recorded in this study, the clerks expressed a greater degree of comfort in the morning than in the afternoon. This is evidenced in Fig. 11, in which it is seen that the curve for the percentage of clerks expressing comfort (*ideal*) at 11 a.m. lies consistently (with one exception) above the corresponding curve for 3 p.m. It
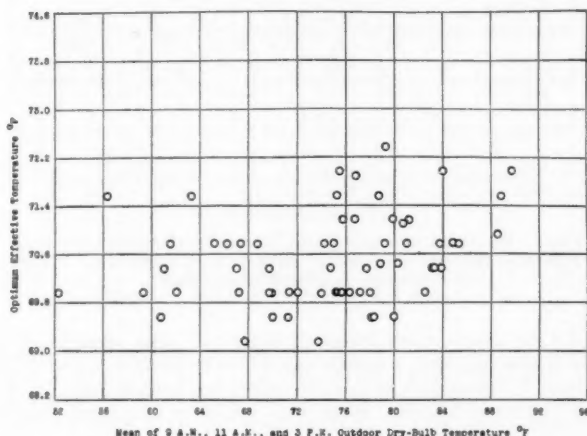
FIG. 6. RELATION BETWEEN DAILY MEAN OF OUTDOOR DRY-BULB TEMPERATURES AT 9 A.M., 11 A.M., 3 P.M. AND OPTIMUM EFFECTIVE TEMPERATURE. JULY, AUGUST AND SEPTEMBER, 1938

will also be seen that the optimum temperature in the morning is in the interval from 70.6 to 70.9 deg ET, which is one degree higher than the optimum for the afternoon, namely, the interval from 69.8 to 70.1 deg ET.

Study of Fig. 12 shows that the curves of percentages of persons reporting a feeling of chilliness in the morning and in the afternoon are hardly distinguishable. On the other hand, the curve for percentage of persons reporting a feeling of warmth in the afternoon lies almost wholly above the corresponding curve for the morning. Thus, the decrease in the feeling of comfort from the morning to the afternoon is accompanied by an increased feeling of warmth.

REACTIONS IN JULY, AUGUST, AND SEPTEMBER COMPARED

The variation in the per cent of votes expressing comfort as indoor effective temperature increases is shown separately for July, August, and September in Fig. 13. Within the range from 70 to 71.8 deg ET, the least comfort was registered in July and the greatest comfort in September, with August falling
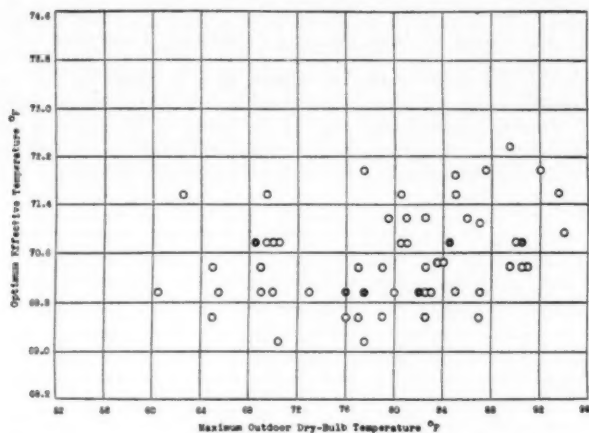
FIG. 7. RELATION BETWEEN DAILY MAXIMUM OUTDOOR DRY-BULB
TEMPERATURE AND OPTIMUM EFFECTIVE TEMPERATURE JULY,
AUGUST AND SEPTEMBER, 1938

in between. At temperatures below 69.6 deg ET, this situation is reversed, for here it is observed that a greater degree of comfort is found in July than in September. Observations in this low range are not available for the month of August. Above 71.8 deg, there was generally a greater expression of
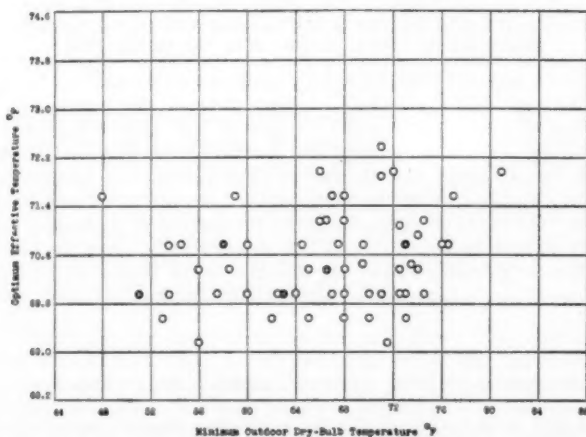
FIG. 8. RELATION BETWEEN DAILY MINIMUM OUTDOOR DRY-BULB
TEMPERATURE AND OPTIMUM EFFECTIVE TEMPERATURE JULY,
AUGUST AND SEPTEMBER, 1938
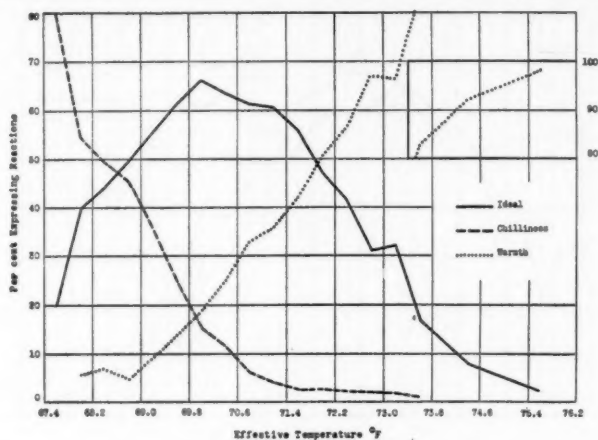
FIG. 9. RELATION BETWEEN EFFECTIVE TEMPERATURE AND PER-
CENTAGE OF PERSONS INDICATING COMFORT, CHILLINESS AND
WARMTH. JULY, AUGUST AND SEPTEMBER, 1938

comfort in September than in August. The sharp upturn in the curve for
July in this high range is without special significance and may arise from
the accidental fluctuations usually ascribed to small numbers of observations.

The optimum temperature for each of three months shows slight variation.
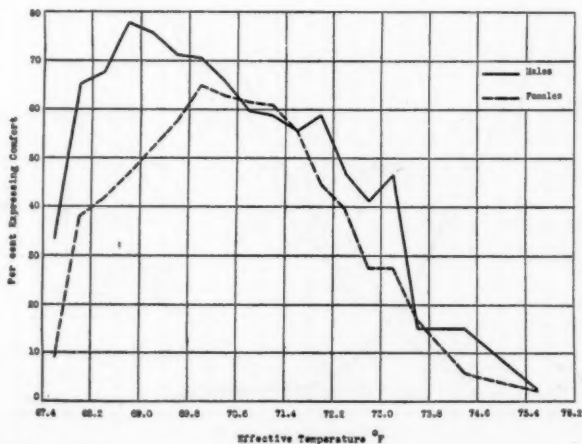
FIG. 10. RELATION BETWEEN EFFECTIVE TEMPERATURE AND PER-
CENTAGE OF PERSONS INDICATING COMFORT FOR MEN AND WOMEN
SEPARATELY. JULY, AUGUST AND SEPTEMBER, 1938
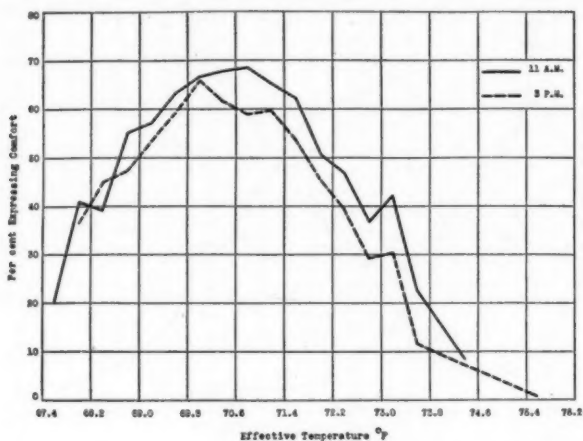
FIG. 11. RELATION BETWEEN EFFECTIVE TEMPERATURE AND PER-
CENTAGE OF PERSONS INDICATING COMFORT AT 11 A.M., 3 P.M.,
BOTH SEXES COMBINED. JULY, AUGUST AND SEPTEMBER, 1938

In July, it was in the interval from 69.8 to 70.1 deg, with 62.1 per cent of
the votes expressing comfort; in August it was in the interval from 69.4 to
69.8 deg (the lowest interval of record for the month), with 70.9 per cent
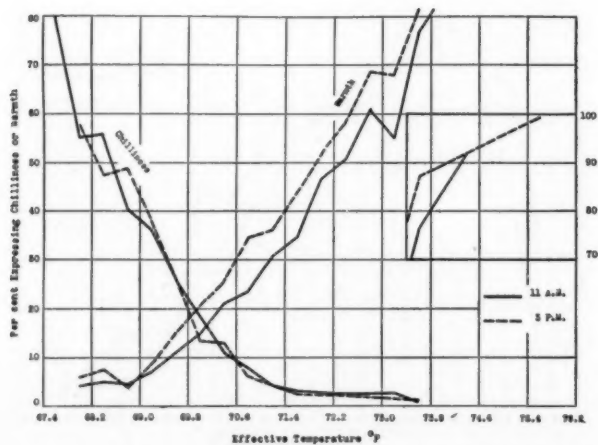
FIG. 12. RELATION BETWEEN EFFECTIVE TEMPERATURE AND PER-
CENTAGE OF PERSONS INDICATING CHILLINESS AND WARMTH AT
11 A.M AND 3 P.M. JULY, AUGUST AND SEPTEMBER, 1938
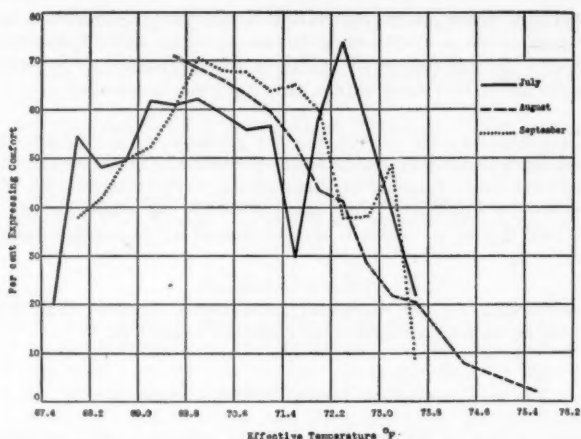
FIG. 13.   RELATION BETWEEN EFFECTIVE TEMPERATURE AND PER-
CENTAGE OF PERSONS EXPRESSING COMFORT DURING JULY, AUGUST
AND SEPTEMBER, 1938.   BOTH SEXES COMBINED

expressing comfort; and in September the optimum interval was again from
69.8 to 70.1 deg with 70.6 per cent expressing comfort.

Within the range of observation, the curve (Fig. 14) for the per cent of
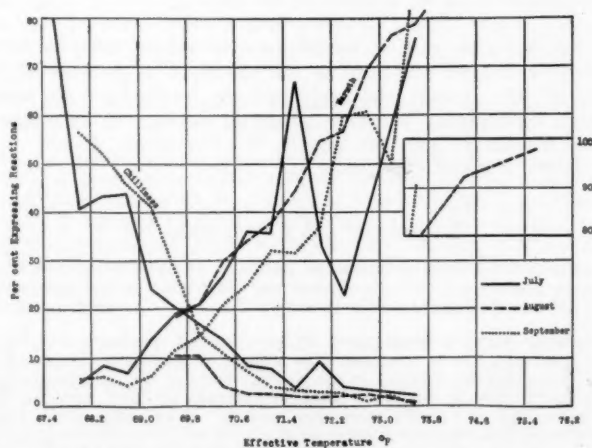
FIG. 14.   RELATION BETWEEN EFFECTIVE TEMPERATURE AND PER-
CENTAGE OF PERSONS INDICATING CHILLINESS AND WARMTH.
JULY, AUGUST AND SEPTEMBER, 1938

votes at various temperatures registering a reaction of *chilliness* in August fell below that of the corresponding curves for July and September. Below 70.0 deg ET, a greater degree of chilliness was registered in September than in July, while above that temperature, this situation is reversed.

Up to 71.2 deg ET, the curves for the percentage of persons expressing a feeling of warmth in July and August are practically identical. At higher temperatures the curve for July becomes erratic, largely because of the chance fluctuation usually associated with small numbers of observations. On the whole, a lesser degree of warmth was expressed in September than in July or August.

The observations in the foregoing paragraphs suggest that the indoor reactions may be related to outdoor temperature conditions.

Comparison with Previous Studies

Laboratory studies in Pittsburgh and Texas pointed to the desirability of a continuous indoor effective temperature of 73 deg as optimum throughout the cooling season. This is somewhat higher than the optimum effective temperature derived from the Ontario study, which was 71 deg. The study made on a group of office workers in Minneapolis indicated that an indoor effective temperature of about 72 deg gave the most satisfactory results in the cooling season from July to the middle of September. In the present study, which is based upon 745 clerks in the new unit of the home office of the Metropolitan Life Insurance Co., the optimum effective temperature is found to be about 70 deg during the months of July, August, and September. This is the lowest optimum effective temperature so far noted.

Where differences in optimum temperatures have been noted in this report, as between men and women, morning and afternoon, or between any two of the months of July, August, and September, the results have not been tested for statistical significance. However, where the findings of the present report confirm the findings of previous reports, the conclusions drawn from them gain the weight of added experience.

Summary and Conclusions

1. The present experiment indicates an optimum effective temperature of about 70 deg during the months of July, August and September in the new unit of the home office of the Metropolitan Life Insurance Co.

2. The optimum effective temperature for men was 1.2 deg below that for women. In general, men are not as susceptible as women to discomfort, especially at the extremes of temperatures recorded in this study. Even at the optimum temperatures for women, which differed somewhat from the optimum for men, men still registered a greater percentage of comfort votes than women. Since most of the home office clerks are women, this observation suggests that summer air-conditioning be regulated to the indoor optimum effective temperature for women, namely, 70 deg.

3. In the morning the optimum effective temperature is about one degree higher than in the afternoon. The decrease in the feeling of comfort from the morning to

the afternoon is accompanied by an increased feeling of warmth. This may perhaps be due to the effect of the mid-day meal.

4. The optimum effective temperatures for July, August and September show slight variation from one month to the other.

5. No significant relationship between outdoor temperatures and indoor optimum effective temperature could be found. The conditions of the experiment were not adapted to determine such correlation, since the range of variation of indoor effective temperatures was limited. Although day-to-day changes in outdoor effective temperatures may be looked upon as a guide to day-to-day changes in indoor effective temperatures, the relation between the two is not so close that it can be depended upon for all occasions.

ACKNOWLEDGMENT

The authors acknowledge their gratitude to Dr. A. J. Lotka, Assistant Statistician of the Metropolitan Life Insurance Co., for his guidance in the analysis of the data and for his help in the preparation of this report, to Mr. F. C. Houghten, Director of Research Laboratory, of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, for giving the benefit of his wide experience in this field of investigation, and to the Bristol Co., for the loan of temperature recording instruments used in this study.

DISCUSSION

A. B. NEWTON (WRITTEN): The authors were able to collect comfort data from a sufficiently large group of subjects so that some very conclusive results are possible and they are to be congratulated on the clear and concise manner in which their conclusions are presented.

It is interesting to note that many of their conclusions agree with the results of the Society's initial work involving large groups of subjects, and that the optimum temperature is very nearly the same for New York City as was shown from the Ontario studies.

The effect of local preferences on the optimum temperature does seem to exist quite definitely but is not as large in magnitude as many people suspect it.

I would agree with the authors that very likely one of the reasons why no relation between daily outdoor temperature and indoor optimum effective temperature was found in their study may be the small range of indoor effective temperatures which was available. I am also wondering whether this effect may partially have been attributable to the fact that the clerks in the New York study do not leave the office, or leave the conditioned space, at noon. In our Minneapolis study, for example, nearly everyone left the conditioned space at noon, either to outdoors or to an unconditioned cafeteria.

The technique of making studies of comfort requirements has became fairly well developed and I think we have reached a point where it is desirable to obtain some accurate data regarding the effect of length of occupancy on the optimum temperature conditions. From observations I have made on actual installations I believe the requirements of varying inside conditions in accordance with outside conditions is greater as the occupancy period becomes less.

Of course the other extreme in occupancy to that covered in the present studies would be studies of restaurant and department store conditions, and some new tech-

nique would undoubtedly have to be developed before such accurate studies could be made.

It is also very desirable that in any studies of this type, regardless of the length of occupancy, they may be extended to cover both the spring and fall changeover period between heating and cooling and cooling and heating cycles. I think it is during these seasons of the year, particularly, that it is most desirable to fluctuate inside conditions in accordance with changes in outside conditions, since by so doing people gradually become acclimated to the weather changes and to the gradual revision of their own feelings.

FEVER THERAPY LOCALLY INDUCED BY CONDITIONED AIR

By Murray B. Ferderber, M.D.,* F. C. Houghten,** M.S., and Carl Gutberlet***
Pittsburgh, Pa.

THE empiric use of heat in the treatment of human ailments has extended for countless years. Among the many forms used may be included packs, steam baths, electrical applications such as radiant energy, short-wave, diathermy, etc.[1] Regardless of the method used the ultimate aim is the dilatation of blood vessels to produce increased circulation which in turn causes the affected part to receive the necessary physiological elements for the eradication of pathology.

In the past several years fever therapy (hyperpyrexia, hyperthermia, etc.) has been used extensively for the elevation of body temperature to destroy an invading organism or to increase the elements of resistance within the body itself.[2] A simple fever therapy apparatus was devised [3] where conditioned air was the means of producing the body temperature desirable for the purpose just mentioned.

However, there are many patients whose physical condition or ailments are not indicated for generalized fever therapy and, up to this time, other methods such as radiant energy and diathermy were used. The physiological changes in circulation are seldom produced within a few minutes, because adjustment to conditions is a property of all life that should bear some consideration. When heat is applied there is an eventual dilatation of blood vessels which

become engorged, and as a rule, an increase in blood flow is desired.[4]   It is believed that this increase, to be effective, should occur over a long time rather than during a few minutes.   Assuming this practice to be sound, it would follow that any apparatus must be designed so as to deliver helpful and comfortable heat without the dangers of burning or shocking.

Following the line of reasoning used in developing the large fever therapy machine,[5] the Department of Industrial Hygiene, School of Medicine, University of Pittsburgh, with the helpful advice of the ASHVE Research Laboratory, developed an apparatus for inducing elevated temperatures in portions of the human body which has proved successful in every respect.   The machine, which may be built to accommodate any portion of the body including

FIG. 1.   PATIENTS BEING TREATED LOCALLY WITH SMALL FEVER
THERAPY UNIT

an upper and lower extremity, hip, back, or shoulder is shown in Fig. 1 designed to accommodate one or both legs.   In this form the device is 43 in. long and 16 in. in diameter and the air conditioning equipment is built in one of the ends of the cylinder while the other end is open to receive the limb.   Like the larger fever therapy box this smaller unit contains an air conditioning system of the general dew-point type in which a highly atomized water spray in the upper end of a small duct supplies both the motive power for circulating air and the heat and moisture for saturating it at the desired temperature.   Fig. 2 shows a more or less ideal arrangement of the duct system in which the downward passage of the spray gives the air a downward momentum removing it from the lower portion of the box and delivering it saturated at the desired elevated temperature into the top of the chamber. The stratified layer at the top of the box is then drawn downward resulting

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[4] Physiological Effects of Heat, by H. C. Bassett. (*American Journal of Physiology*, LXXIII: 127, 1933.   Handbook of Physical Therapy *AMA*, 2nd Ed.).
[5] Loc. Cit. Note 3.

in little or no temperature gradient throughout a horizontal cross-section with a minimum controlled gradient from the top to the bottom.

With a 40-lb water pressure and a 4-in. duct throughout the air conditioning system 9.8 cu ft of air per minute is removed from the bottom of the box, heated and saturated to the desired temperature, and returned to the upper portion. Without insulation 20 to 30 gal of water per hour are sufficient for satisfactory conditions in the box, and if the cylinder is insulated to
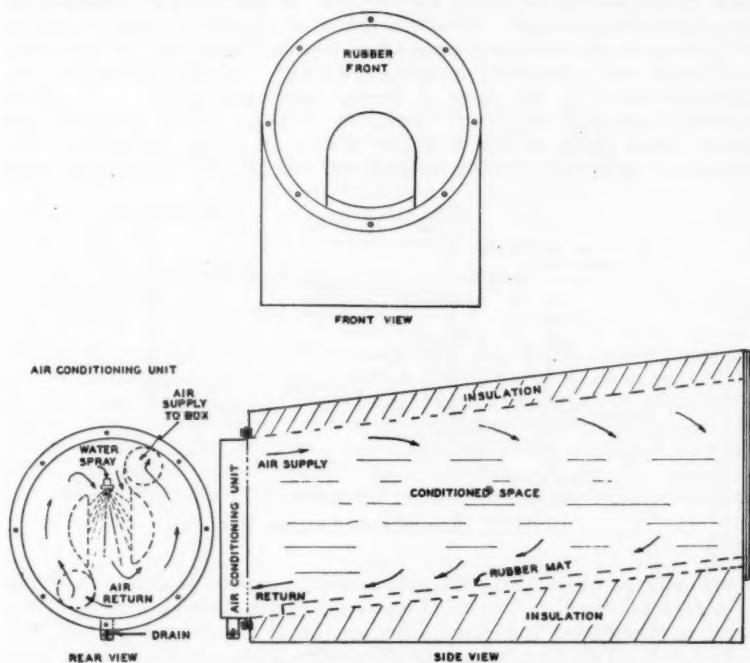
FIG. 2. LOCAL FEVER THERAPY BOX WITH STRATIFIED CIRCULATION

insure against excessive heat loss to the surrounding atmosphere, a considerably smaller water volume will suffice. A water temperature of 130 F will produce a uniform saturated atmosphere of from 120 to 125 F. Since the air is saturated upon entering the box and loses heat before being returned to the air conditioning part of the cycle, saturation is insured throughout.

Fig. 3 shows a somewhat simpler arrangement, which although lacking the stratifying characteristic in Fig. 2 and insuring a less perfect saturated cycle, gives sufficiently satisfactory performance in such a small apparatus for ordinary purposes. Fig. 4 shows the relation between the pulse before and after treatment, while Fig. 5 shows the change in leucocyte count.

From a comparative standpoint the whirlpool baths using rapidly agitated water from about 108 to 120 F have been used in industrial hospitals to

produce a hyperemia (increased blood supply). The upper range of temperature in this water bath is apparently too great for comfort and could conceivably cause certain difficulties which might rise in a patient with impaired circulation in the extremities. The ability of a patient to tolerate temperatures between 114 and 120 F, saturated, is outstanding since most of the cases are treated at these temperatures without danger of complications. From experience it has been found that it is necessary to treat a patient 1 to 3 hours to produce desired effects, and the index of improvement increases with each succeeding treatment. Usually a course of treatments would consist of 36 hours at a pre-determined temperature usually divided into 12 treatments of 3 hours each given daily or every other day. At this writing the only contra-indications to this form of therapy have been sloughing of a part including gangrene, very severe circulatory deficiency in the extremity, and trophic ulcers which are due to certain diseases of the nervous system. The increase in pulse rate is fairly insignificant, and the rise in leucocyte count
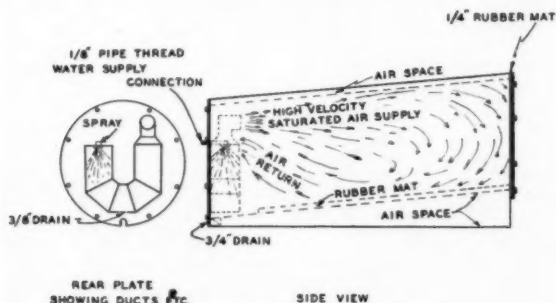
Fig. 3. Local Fever Therapy Box

would average about 1500 cells, which may be purely a personal variation in making the counts. The rise in body temperature resulting from this treatment is insignificant and varies from no rise to a maximum rise of from 1 to 1½ F. This increase did not affect even those patients suffering from severe cardiac lesions.

Most of the patients enjoy the treatment because the apparatus is placed on a comfortable hospital bed, and frequently the individual falls fast asleep without the addition of a sedative. The observations indicate that men tolerate temperatures as high as 125 F, while women are more comfortable between 115 and 118 F. In attempting to evaluate its clinical value the cooperation of a prominent industrial surgeon was fortunately obtained, whose wide experience would add much to the experimental knowledge.[6] The first patients treated were those who had sustained fractures following which stiffness of surrounding joints and tendons resulted in disability. Most of these individuals were treated with other physical methods, but there was little or no improvement. To the satisfaction of all workers improvement was noticeable follow-

[6] Local Application of Fever Therapy, by M. B. Ferderber, J. Huber Wagner, A. Sherrill and Samuel Sherman. (Preliminary Report, *Departmental Publication.*)
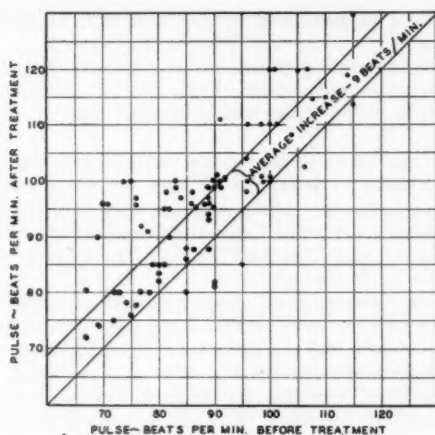
FIG. 4.   RELATION BETWEEN THE PULSE BEFORE
AND AFTER TREATMENT

ing the inception of saturated heat treatments.   In some few cases where the
stiffness persisted, gentle manipulations were performed by the surgeon followed
by these heat treatments, and improvement became more evident.   Very often
following open reduction of fractures, or from other causes, edema (swelling)
of an extremity may occur, characterized by considerable enlargement of the
leg or arm and a shiny, stretched skin.   Local fever therapy was especially
efficacious in this type of disability, and both lymphatic and blood circulations
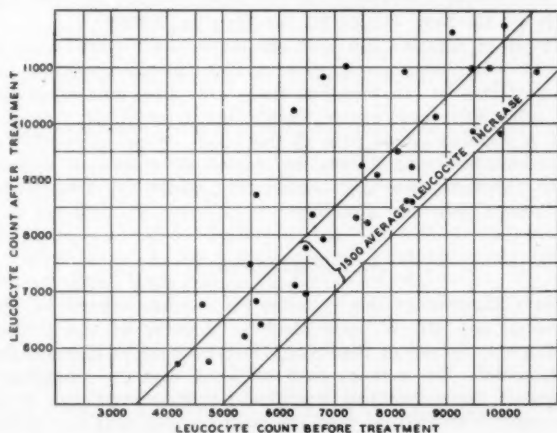
FIG. 5. CHANGE IN LEUCOCYTE COUNT BEFORE AND AFTER
TREATMENT

were vastly improved as evidenced by the reduction of swelling previously mentioned.

The so-called simple sprain produces rather extensive disability and *time off* whether it be industrial or domestic. Early intensive treatment is indicated, and almost immediate relief from this particular type of accident has been obtained.

In the course of the investigations many patients with varicose ulcers were treated routinely in order to determine the uses for which this modality might be indicated. The satisfactory results were not only surprising but gratifying, and it has since become a routine treatment in one of the hospitals cooperating with the Fever Therapy Research Fund of the Department of Industrial Hygiene, University of Pittsburgh.

Peripheral vascular diseases result from impaired circulation of the extremities, and it is axiomatic in medicine that a suitable form of heat is necessary to aid in the relief of pain and the restoration of collateral circulation. While experimental evidence would indicate the value of moist heat, additional work is necessary to definitely establish the range of temperatures and the time allotted for treatment. It is known, however, that lower dry-bulb temperatures are necessary in some cases, and good clinical judgment should dictate that this form of treatment must not produce pain in these patients. Animal experiments carried on in the laboratories of the Fever Therapy Research Fund definitely show that the rate of blood flow is considerably greater when moist heat is used, and the deep muscle temperatures produced show a maximum of 107 F, which is definitely in the safe physiologic range. However, another method for inducing local heat causes a deep temperature of approximately 109 F, and there is the possibility that this temperature may be either above. or near the upper limit of physiological tolerance. Another very interesting observation was the retention of the deep muscle heat which lasted considerably longer when produced by saturated heat. The value here lies in the fact that the patient receives the advantage of continued dilation and increased blood supply which are the basic causes for the deep temperature.

The treatment of arthritis has been subject to much controversy for many years and aside from those cases resulting from specific infection, there have been as many forms of treatment as there are physicians recommending it. We do not offer this form of treatment as a remedy for this disease, but where there is localization of pain in the joints of an extremity, relief has been given so that following the original course of treatments, patients remain symptom-free for some time and occasionally return for treatment to avoid complete return of pain and disability.

AIR FLOW MEASUREMENTS AT INTAKE AND DISCHARGE OPENINGS AND GRILLES

By G. L. Tuve* and D. K. Wright, Jr.,** Cleveland, Ohio

SUMMARY

TO OBTAIN true air velocity or volume from the readings of air velocity meters, an *application factor* must be applied in addition to the corrections for instrument calibration.

For rotating vane-wheel anemometers, the recommended application factors for most conditions are 0.85 for air inlet openings, and 1.03 for air discharge openings. For the velometer used at air discharge openings an approximate application factor of 0.93 is recommended. The limitations of these factors are given in Tables 1 and 2. These two tables are the result of correlations between the 1000 or more instrument traverses here reported and the data presented elsewhere by several other investigators.

The influence of each of six important variables upon instrument readings is discussed, for both air intake and air discharge openings.

DEFINITIONS: TRUE VELOCITY *vs* INSTRUMENT READING

The air velocity across the face of an opening is not uniform, in fact a wide range of velocity readings can frequently be obtained.[1] Since an air intake or an air discharge opening is usually a contracted section, the velocity varies both at right angles to the flow and in line with it. A velocity reading refers to a single location, and the reading will probably change if the instrument is moved either from side to side or in or out.

But there are no instruments for measuring air velocity at a *point,* because an instrument, tube or jet will occupy an appreciable area across the face. The instrument will also be affected by other conditions, such as pressure or

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  [1] See Fig. 10.

TABLE 1—APPLICATION FACTORS FOR GENERAL USE OF ANEMOMETERS
AND VELOMETERS

	APPLICATION FACTORS	
	Anemometer 3 In. to 6 In. Sizes Velocity 400–1500	Velometer Spot Jet or Averaging Jet Velocity 600–1500
AIR INTAKE OPENINGS (ROOM EXHAUST) Any size or shape of rectangular intake more than 4 in. wide and up to 600 sq. in. with flange at least 2 in. wide either free-open or with grille with free opening 60 per cent or more of the core area:	0.85	Not recommended
AIR DISCHARGE OPENINGS (ROOM SUPPLY) Any size or shape of rectangular discharge opening more than 4 in. wide and up to 600 sq. in. area, with length of approach duct at least equal to twice the smaller side, and with free opening 70 per cent or more of the core area, no directional vanes:	1.03	0.93

EQUATION FOR USE:

$$\text{True Volume, cfm} = \left\{\begin{matrix}\text{Application}\\\text{Factor}\end{matrix}\right\} \times \left\{\begin{matrix}\text{Av. Velocity}\\\text{by Instrument}\end{matrix}\right\} \times \left\{\begin{matrix}\text{Designated}\\\text{Area}\end{matrix}\right\}$$

Designated Area for intakes = Core area.
Designated Area for discharge openings with anemometer = average between core area and free-open area at the plane of the near face.
Designated Area for discharge openings with velometer = free-open area.
Average velocity by instrument is obtained from equal-time traverse of 4 in. squares, corrected for instrument calibration.
Note: For data on accuracy of these application factors, see text.

temperature, as well as by velocity. Hence the velocity pattern obtained will depend on what instrument is used. If an average velocity is required, an arbitrary method of averaging must be agreed upon. If the average velocity is to be multiplied by an area to obtain volume, that area must be clearly defined or designated.

The following statements and definitions are in accord with the general practices of many test engineers, and will be used for the purposes of this investigation:

Location of Instrument: If the opening is covered with a grille, the instrument should touch the grille face, but should not be pushed in between the bars. For a free opening without a grille, guide wires should be stretched across the plane of the opening and the instrument held in the plane of the guide wires. The anemometer must always be held in such a manner that the air flow through the instrument is in the same direction as was used for calibration (usually from the back toward the dial face). An instrument, tube or jet should be held in place by means of a thin handle (½ in. in diameter or less) and the hands and body of the observer should be entirely outside the area of flow. A string and rubber band may be used for starting and stopping the totalizer.

Average Velocity by Instrument: The total rectangular opening or core area is to be divided into squares or rectangles 3 in. to 5 in. on a side, and the average velocity by instrument will be the arithmetical average of the readings in these

TABLE 2—CORRECTION FACTORS

For low air velocities or very large openings, the instrument application factors given in Table 1 should be multiplied by the following additional correction factors.

AIR VELOCITY CORRECTION

VELOCITY BY INSTRUMENT, FPM	ROTATING VANE ANEMOMETER		VELOMETER
	Factor for Intake	Factor for Discharge	Factor for Discharge
700	1.00	1.00	1.02
600	1.00	1.00	1.05
500	1.00	1.01	1.09
400	1.01	1.03	1.15
300	1.03	1.06	1.22
200	1.07	1.10	....

SIZE CORRECTION
(*For Intakes*)

AREA OF INTAKE OPENING, SQ IN.	CORRECTION FACTOR
500	1.01
1000	1.07
1500	1.14
2000	1.16

Caution Approach Condition:
The factors given in Tables 1 and 2 *do not* apply to duct-end intakes without flanges or to discharge openings in thin-walled plenum chambers.

squares. Totalizing instruments may be moved from one square to another without stopping the totalizer, but an interval of at least 10 seconds in each square is recommended.

True Average Velocity: The true average velocity is obtained by dividing the true volume of air in cfm by the designated area in square feet. (In calibration tests the true volume of air is measured by an independent meter of approved accuracy.) The true average velocity is also obtained as the product of the application factor and the average velocity by instrument.

Designated Area: (a) For air *intake* openings, the designated area is the core area, or the total area within a rectangle drawn through the outer edges of the outer openings. (b) For air *discharge* or outlet openings, the designated area may be either the free area or the arithmetical average of the core area and the total free-open area at the plane of the near face, depending on the instrument used, as in Table 1.

Application Factor: The application factor is the factor by which the average velocity by instrument is to be multiplied to obtain the true average velocity.

Equations:

$$\left(\begin{matrix} \text{Average velocity by} \\ \text{instrument, fpm.} \end{matrix} \right) \times \left(\begin{matrix} \text{Application} \\ \text{Factor} \end{matrix} \right) = \left(\begin{matrix} \text{True Average} \\ \text{Velocity} \end{matrix} \right)$$

$$\left(\begin{matrix} \text{True average} \\ \text{velocity, fpm.} \end{matrix} \right) \times \left(\begin{matrix} \text{Designated} \\ \text{area, sq ft} \end{matrix} \right) = \left(\begin{matrix} \text{True} \\ \text{Volume cfm.} \end{matrix} \right)$$

Instrument Calibrations and Test Methods

The velocity instruments (anemometers and velometers) used in this investigation were new instruments, calibrated before and after the tests by the methods described in a previous paper.[2] To obtain instrument readings, the calibration corrections were applied in all cases. Calibrations in a wind tunnel were taken as standard for the velometers, and calibrations by a 10-ft
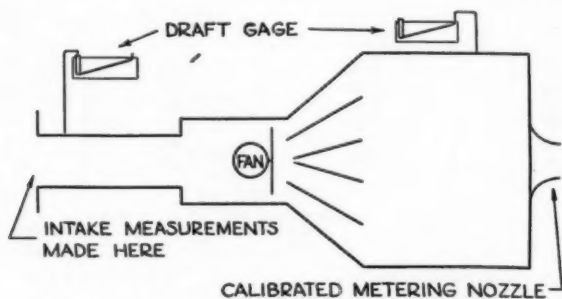
Fig. 1. Diagram of Test Unit for Air Intake Studies
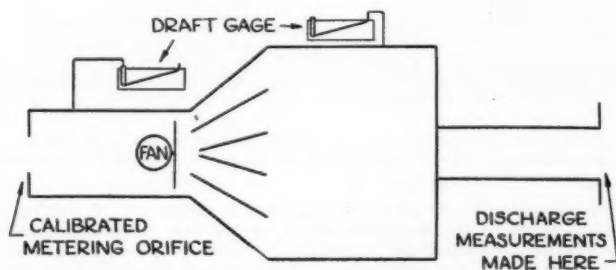
Fig. 2. Test Set-up for Study of Air Discharge Openings

rotating arm were used for the anemometers. With an arm length of 10 ft or more, the agreement between the two methods is within about 2 per cent.

The test equipment for air intakes is shown in Fig. 1 and that for air discharge openings is shown in Fig. 2. The primary air measurement was by calibrated nozzles and orifices, which had been checked against each other and against impact-tube and pitot-tube traverses with an accuracy closer than 1 per cent. Inclined manometer gages, calibrated within 0.001 in. of water were used for measuring impact pressures and differential pressures, the calibration standard being the water-filled precision hook gage shown in Fig. 3. This gage could be read to the nearest 0.0005 in. of water. Plenum and duct pressures were always measured by two independent gages, each connected to

[2] ASHVE Research Report No. 1140—The Use of Air Velocity Meters, by G. L. Tuve, D. K. Wright, Jr., and L. J. Seigel. (ASHVE Transactions, Vol. 45, 1939, p. 645.)

a different set of pressure taps. All ducts and plenum boxes were carefully sealed, and were calibrated for leakage both before and after the tests.

The question of what test methods to use in the determination of instrument application factors requires some analysis of stream characteristics as well as of instruments. When an air stream flows through a contracted section such as an orifice, a nozzle or a grille, the drop in static pressure and the increase in velocity do not take place instantaneously. On the contrary, an appreciable length or distance in the direction of flow is required for the minimum down-stream pressure to be reached and the maximum downstream velocity attained.
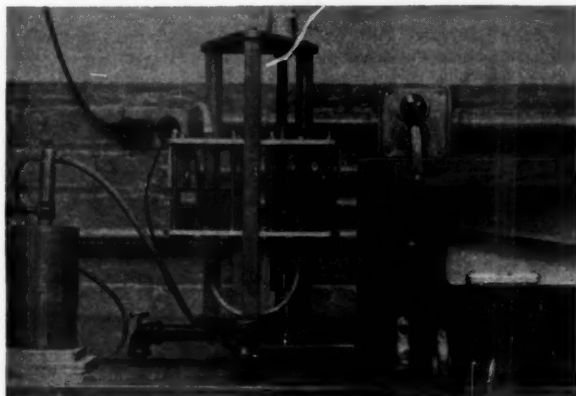
Fig. 3. Hook-gage Micromanometer Used for Draft Gage
Calibrations

Similarly, the velocity distribution across the stream varies with the type and length of approach.

Apparently then, none of the common instruments (except a modified pitot tube) will measure true velocity, because the pressure gradients and the velocity gradients are not properly accounted for. The instrument application factor must therefore compensate for these gradients, as well as for the defects inherent in the instrument when it is used in non-uniform streams. And finally, this application factor must be based on rigid and arbitrary methods of calibrating and using the instrument, especially if average velocity is required. These difficulties lead to the simple definition of the application factor as the true metered velocity divided by the instrument reading (or the average of several readings). For rectangular openings, which are the only ones considered in this paper, the area is arbitrarily divided into small squares or rectangles 3 in. to 5 in. on a side, and a reading is taken in each. The instrument calibration is then applied.

If the rotating arm method can be used for calibrations, it is by far the easiest and most consistent for velocities below 1000 fpm. If a wind tunnel

is to be used, it must have a long smooth inlet nozzle, with supply from a very large duct or plenum or from the atmosphere in a quiet room. The readings must be taken far enough downstream to obtain the maximum velocity. Under no circumstances should air velocity meters be calibrated in a fan duct, or in the discharge from a duct or pipe, or from a square-edged orifice or a short-type nozzle. There is no satisfactory way of checking the errors produced in these latter cases.

Variables Affecting Instrument Readings

Six major variables affect the reading of a velocity meter when it is applied to either an air intake or an air discharge opening:

1. Type of approach to the opening.
2. Total area of opening.
3. Shape of opening, or dimensional ratio.
4. Air velocity.
5. Type, size and position of instrument.
6. Screen or grille, if any.

Each of these six factors has been investigated in turn, first for air intakes, then for air discharge openings.

Air Intake Openings

Room Exhaust

All intake openings move air by a conversion of static energy to velocity energy. Hence a suction or vacuum exists in or near an air intake, and the impact-type of measuring instrument such as an impact tube or a velometer jet reads *zero* when placed in an intake opening facing against the stream. The reversed tube, facing downstream, is unsatisfactory because of the shape-factor. The common pitot tube is also unsatisfactory, because its impact and static openings are too far apart and because low-velocity measurements are very difficult. The rotating vane-wheel anemometer reads too high in most cases, but its readings are highly consistent and it is the most convenient instrument to use. An investigation of the use of the anemometer at intake openings has therefore been made.

An air intake opening in a wall is essentially the upstream side of a flat-plate orifice, or of many orifices side by side. The jet contraction which takes place with flow through such an orifice affects the readings of an anemometer traverse. If the air intake is at the end of a thin-walled duct, with no flange at the entrance, the flow pattern will be entirely different, and hence a different anemometer application factor must be used. An intake opening with a flange is intermediate between a wall opening and an open-end duct, but unless the flange is very narrow it acts like a wall opening, as is demonstrated by the test data of Fig. 4 (see also Table 3).

The shape and size of an air intake opening has less effect than might be expected. If the opening is over 4 in. wide and not more than 400 sq in. in area, the anemometer application factor remains constant at about 0.85. Examples from tests are given in the first section of Table 3. For intake

openings of 500 sq in. or more, the application factor increases due to less *edge effect* (see Fig. 10), and a correction factor from Table 2 should be applied.

In the velocity range from 400 to 1500 fpm the anemometer application factor is practically constant, as is shown by the typical traverses of Fig. 5, as well as by the deviations listed in Table 3 covering about 500 traverses. For air velocities below 500 fpm the application factor increases, and a correction must be applied as given in Table 2.

The size of anemometer has little effect as long as the traverse squares are nearly the same size as the anemometer diameter. Results of tests in a 12-in. square intake, with three sizes of anemometers and with 1 to 16 traverse positions are shown in Fig. 6.

Some anemometers have a convex cover-glass over the dials which will not allow the shroud ring to touch the grille face. These instruments are not properly designed for air intake measurements and errors of the order of

8 in. Flange No Error	4 in. Flange No Error	2 in. Flange No Error	1 in. Flange Error Less than 1 per cent	No Flange Error 15 per cent
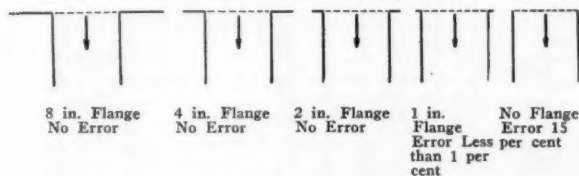
FIG. 4. TEST RESULTS SHOWING EFFECT OF WIDTH OF FLANGE
ON ACCURACY OF ANEMOMETER FACTOR RECOMMENDED IN TABLE 1.
TESTS ON 12-IN. SQUARE INTAKE

5 per cent are produced. But accurate readings may be obtained by fitting an extension on the shroud ring to bring it flush with the cover glass. (An extension may be improvised by using paper masking tape.)

When screens or low-resistance grilles are used to cover an air intake, the anemometer application factor is practically the same as for a free intake. Tests were made on various sizes of intake grilles and screens, including a square-punched grille of 71 per cent free area, three or four types of bar grilles having 75 per cent to 90 per cent free area and a ½-in. mesh wire screen of about 85 per cent free area. In almost all cases the anemometer application factors were within 3 per cent of the selected factor of 0.85 (see Table 3).

AIR DISCHARGE OPENINGS

Room Supply

The problem of the air discharge opening is more complicated than that of the air intake for three reasons: (1) More types of instruments are available for air discharge measurements. (2) Conditions of the approach to an air outlet are highly variable. (3) Many kinds of grilles and registers are in common use.

In the matter of instruments, the present investigation has been confined to the anemometer and the velometer, which are the two most common. It is

TABLE 3—DATA ON ANEMOMETER MEASUREMENTS AT AIR INTAKES

Size of Opening	Type of Approach	Size of Anemom. In. Dia.	No. of Traverse Positions	No. of Complete Traverses	No. of Test Veloci-ties	Velocity Range, Fpm	Average Anemom. Appli-cation Factor, F	Percentage Deviation From Avg. F	
								Mean	Max.
EFFECT OF SIZE OF OPENING									
14x5	Wide Flange	4	4	15	5	435–1215	0.854	0.5	1.0
14x6	Wide Flange	4	8	12	4	610–1205	0.840	0.1	0.5
14x8	Wide Flange	4	8	23	7	440–1350	0.856	2.2	3.3
14x10	Wide Flange	4	12	24	8	445–945	0.860	0.5	1.5
24x5	Wide Flange	4	6	20	6	595–1230	0.864	0.7	1.0
8½x8½	Wide Flange	4	4	19	6	565–1360	0.856	0.8	1.6
12x12	Wide Flange	4	9	28	9	444–932	0.834	0.4	1.0
18x18	Wide Flange	4	16	3	1	274	0.864	..	..
24x24	Wide Flange	4	36	9	3	319–370	0.880	0.4	0.6
30x42	Wide Flange	4	80	7	3	460–580	0.973	0.9	1.3
36x50	Wide Flange	4	70	10	2	575–630	0.995	..	0.5
EFFECT OF FLANGE									
12x12	8″ Flange	3	16	12	4	470–935	0.834	0.2	0.6
12x12	4″ Flange	4	9	28	9	444–932	0.834	0.4	1.0
12x12	2″ Flange	4	9	25	8	336–933	0.837	0.7	2.1
12x12	1″ Flange	4	9	25	7	330–940	0.840	1.3	2.9
12x12	Duct End—No Flange	4	9	16	5	313–893	0.716	0.5	0.8
EFFECT OF LOW VELOCITIES									
24x24	Wide Flange	4	36	3	1	130	0.985	..	0.0
24x24	Wide Flange	4	36	3	1	178	0.938	..	0.2
24x24	Wide Flange	4	36	3	1	228	0.910	..	0.0
24x24	Wide Flange	4	36	3	1	267	0.895	..	0.0
24x24	Wide Flange	4	36	3	1	319	0.885	..	1.1
24x24	Wide Flange	4	36	3	1	352	0.880	..	0.5
24x24	Wide Flange	4	36	3	1	370	0.880	..	0.0
EFFECT OF NO. OF TRAVERSE POSITIONS									
12x12	Duct End—No Flange	3	1	18	6	340–995	0.827	0.8	1.4
12x12	Duct End—No Flange	3	4	21	6	450–955	0.743	0.1	0.4
12x12	Duct End—No Flange	3	9	29	9	304–985	0.717	1.4	4.2
12x12	Duct End—No Flange	3	16	43	15	316–954	0.719	0.9	2.5
12x12	Wide Flange	3	4	12	4	440–955	0.879	0.6	1.0
12x12	Wide Flange	3	16	12	4	470–935	0.834	0.2	0.6
EFFECT OF ANEMOMETER SIZE									
12x12	Duct End—No Flange	6	4	29	8	390–994	0.723	0.6	1.5
12x12	Duct End—No Flange	4	9	16	5	313–893	0.716	0.5	0.8
12x12	Duct End—No Flange	3	16	43	15	316–954	0.719	0.9	2.5
12x12	Wide Flange	4	9	28	9	444–932	0.834	0.4	1.0
12x12	Wide Flange	3	16	15	5	357–935	0.837	0.7	1.8
24x8	Flange and Grille	3	12	6	2	399–839	0.876	..	1.1
24x8	Flange and Grille	4	12	6	2	618–1000	0.888	..	0.2
EFFECT OF GRILLE									
24x8	Bar Grille and Flange	3 and 4	12	12	4	400–1000	0.882	0.6	1.6
20x8	Sq Punched Grille	4	10	14	4	540–1460	0.780	0.5	2.0
12x6	Sq Punched Grille 71% Free Area	4	6	15	5	700–1050	0.874	0.8	1.6
12x6	Bar Grille and Flange	4	6	33	11	640–1080	0.856	1.6	3.4
14x10	Screen—⅛″ Mesh—85% Free Area	4	12	15	5	660–1210	0.857	0.6	1.2

admitted that one or more of the heated type instruments might be used, but some of these require more time for the taking of readings, and the first cost of the quick-reading types is rather high. Tests were made with three kata-thermometers, at air velocities from 200 to 2000 fpm, but, due to inconvenience in reading and to uncertainty in calibration, the use of these instruments was discontinued.

Using the anemometer and the velometer, the position and method of traverse were again arbitrarily fixed, by using small (3 in. to 5 in.) squares or rec-
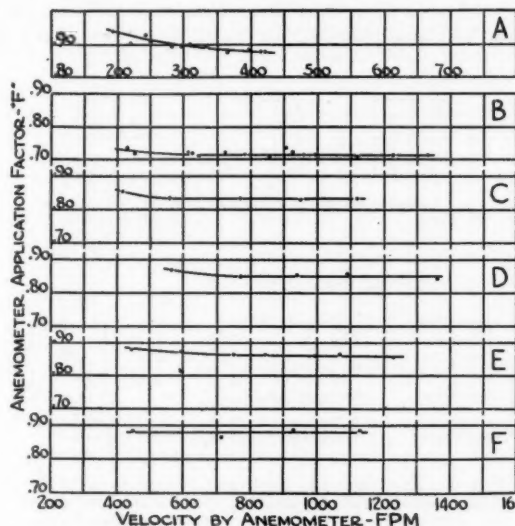
FIG. 5. TYPICAL CURVES OF APPLICATION FACTOR *vs* AIR
VELOCITY FOR AIR INTAKES

A—24 in. sq flanged open duct   D—8½ in. sq flanged open duct
B—12 in. sq sharp-edged open duct   E—5 in. x 24 in. flanged open duct
C—12 in. sq flanged open duct   F—8 in. x 24 in. low resistance grille

tangles and taking a reading at the center of each. Any size of anemometer, 3-in. to 6-in., and either the spot jet or the averaging jet for the velometer, may be used in this traverse method with practically the same results.

The approach condition upstream from the opening is the variable having the greatest effect upon the instrument factor for air discharge or room supply openings. In fact, if approach conditions cannot be controlled, the air measurements by traverse with a velocity-type meter are only roughly approximate.

The two extreme conditions of the approach are: (1) A very long, straight approach duct, and (2) a large thin-walled plenum chamber with a square-edged opening. In the first case the approach velocity is a maximum; in the second case the approach velocity is zero and all the velocity is generated at the opening itself. When either an anemometer or velometer is used to

traverse such a square-edge opening in a plenum or large duct the instrument reading will be high, and an application factor of 0.65 to 0.75 must be applied to obtain the average velocity. The use of an application factor in this case is rather unsatisfactory, because it is greatly affected by edge conditions and by the presence of even a narrow flange or frame.

For a duct-type approach approximately the same size as the opening, the application factor is near unity for the anemometer and from 0.87 to 1.01 for the velometer. Tests were made to determine the necessary length of approach duct to produce a constant application factor for the instruments. Starting with a 12 in. x 12 in. duct 67 in. long, this duct was cut off 7 times, until only a 1-in. long stub remained for the 8th test. The results are shown in

3 in. Anemometer   4 in. Anemometer   6 in. Anemometer
No Error   No Error   Error less than
                                      1 per cent

16 squares   9 Squares   4 Squares   Center Only
No Error   No Error   Error   Error
                           4 per cent   12 per cent
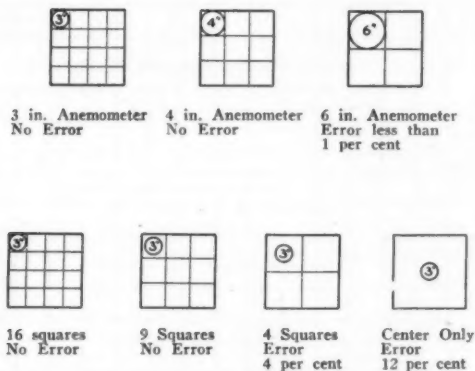
Fig. 6.—Test Results Showing Effect of Size of
Anemometer and Number of Traverse Positions
on Accuracy of Anemometer factor Recommended
in Table 1. Tests on 12-in. Square Intake

Fig. 7 and in Tables 4 and 5. In this case a length of straight approach duct of 1.5 diameters or more was necessary to produce a consistent application factor.

The effect of variations in the air velocity upon the instrument application factor is negligible as long as the velocity is within the range of 500 to 1500 fpm for the anemometer or 700 to 1500 fpm for the velometer. This is shown by the typical test curves of Figs. 8 and 9. For air velocities lower than these, the application factors increase, and corrections must be made as given in Table 2. These correction factors were obtained by replotting a large number of test curves like those in Figs. 8 and 9.

The size and shape of the openings have little effect on the instrument application factor for air discharge or supply openings, as long as the width is 4 in. or more and the area does not exceed 600 sq in. No tests were made on discharge openings larger than 24 in. x 24 in.

The effect of grilles covering air discharge or supply openings is to make the instruments read low if the calculations are based on the free area only.

The method of using the average between the free area and the core area, which was originated by Davies[8] and is recommended by THE GUIDE 1940 (p. 770), gives instrument application factors near unity for the anemometer. For the velometer on the other hand, the advantage of using the average is questionable (see Tables 4 and 5).

CONCLUSIONS AND CORRELATIONS

The general conclusions from these studies are that the rotating vane anemometer can be relied upon for air volume measurements with an accuracy of 3 per cent or closer, if the measurements are made at a free-open intake (with flange) or at a free-open discharge from a long duct.   When the opening is
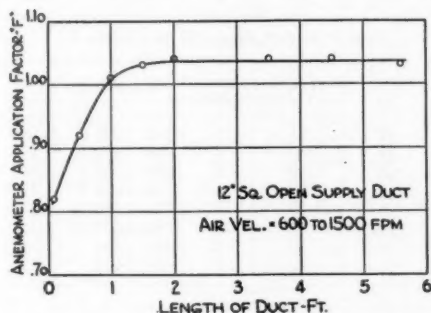
FIG. 7.   EFFECT OF LENGTH OF APPROACH TO
AIR DISCHARGE OPENINGS

Note: Each point represents
12 traverses at 4 air velocities.

covered with a grille or screen the results are not as reliable, but an accuracy of 5 per cent can be expected, both for flanged intakes and for discharge or supply openings which have an approach duct at least 2 diameters in length. The instrument factor is little affected by the area or the shape of the opening, by the instrument size or by the air velocity (above 500 fpm).   For comparative measurements the anemometer is highly accurate,—within 1 per cent or less.

Perhaps the most important result of these tests has been to demonstrate that errors as great as 15 per cent to 30 per cent may be caused by neglecting the effect of approach conditions.   The anemometer factor for an air intake may change from 0.85 to 0.72 if the flange is omitted.   The factor for both anemometer and velometer, applying to an air discharge grille, may be reduced 15 to 30 per cent if the grille is moved from a location at the end of a long duct to a location in the wall of a plenum chamber.

The velometer is an excellent instrument for instantaneous readings, but the

[8] ASHVE RESEARCH REPORT No. 857—The Measurement of the Flow of Air Through Registers and Grilles, by L. E. Davies.   (ASHVE TRANSACTIONS, Vol. 36, 1930, p. 201.)   (See also RESEARCH REPORT No. 911, Vol. 37, 1931 and RESEARCH REPORT No. 966, Vol. 39, 1933.)

Table 4—Data on Anemometer Measurements at Air Supply Openings

Size of Opening	Type of Approach	No. of Traverse Positions	No. of Complete Traverses	No. of Test Veloci-ties	Velocity Range Fpm	Average Anemom. Applica-tion Factor F	Percentage Deviation From Avg F	
							Mean	Max.

Effect of Size of Free Openings

14x5		8	31	10	450–1100	1.04	1.1	3.4
14x8	24″ long	8	24	8	600–1500	0.99	0.6	1.0
14x10	Straight	12	12	4	650–1200	1.02	0.9	2.7
12x12	Duct	16	12	4	550–1400	1.04	0.5	1.9
24x8	8′ long	16	9	3	600–1200	1.03	0.2	0.3
24x24		64	12	4	500–700	1.02	0.4	1.0

Effect of Low Velocity—See Table 2.

Effect of Length of Approach
(Free Openings)

	1″ long	16	15	5	500–1400	0.82	0.4	0.6
	6″ long	16	12	4	600–1400	0.92	0.7	1.1
12x12	12″ long	16	12	4	550–1200	1.01	0.9	1.5
Straight	18″ long	16	12	4	600–1400	1.03	0.3	0.5
Open	24″ long	16	12	4	550–1400	1.04	0.5	1.9
Duct	42″ long	16	12	4	550–1400	1.04	1.0	2.0
	52″ long	16	12	4	550–1400	1.04	0.9	1.3
	67″ long	16	12	4	600–1400	1.03	0.6	1.3

Effect of Grilles							F Based on Mean Area
20″x8″ Sq. Punched 71% Free Area	10	15	5	500–1400	1.19	1.7	2.5	0.99
24″x8″ V-Bar 76% Free Area	14	12	4	550–1400	1.19	0.4	1.0	1.02
24″x8″ H-Bar 85% Free Area	16	12	4	500–1300	1.17	0.7	0.8	1.08
12″x6″ H-Bar 78% Free Area	9	12	4	300–1300	1.14	..	..	1.01

(Approach to Grilles: Nominal Size Rectangular Duct—2 ft long.)

Tests on Plenum Openings

18x6 Plenum	10	15	5	300–1600	0.67	...
12x6 Plenum	6	15	5	700–1700	0.73	...
12x6 Bar Grille ½″ centers, 78% F.A.	6	12	4	250–500	1.07	0.95
18x6 Bar Grille 79% F. A.	10	12	4	600–1100	1.03	0.92
20x8 Bar Grille 79% F. A.	12	27	9	250–800	1.02	0.91
24x8 Bar Grille ¼″ centers, 76% F. A.	16	33	11	400–1400	1.01	0.88

fluctuations inherent in a turbulent air stream make it difficult to determine the average velocity by the use of any instrument of the instantaneous indicating type. Snap readings in the center of an air supply opening can probably be relied upon just as closely as the traverse readings, if the velocity distribution across the opening is good. Table 5 shows that a factor of 0.86 applied to the

TABLE 5—DATA ON VELOMETER MEASUREMENTS AT AIR SUPPLY OPENINGS

Size of Opening	Type of Approach	Type of Jet Used	No. of Traverse Positions	No. of Test Veloci- ties	Velocity Range Fpm	Aver- age Velom. Appli- cation Factor, F	Percentage Deviation From Avg F		Factor F for Center Read- ing Only
							Mean	Max.	
			EFFECT OF SIZE OF FREE OPENINGS						
14x5	24 in.	Averaging	4	7	800–1600	0.92	1.5	4.0	0.90
14x8	Long	Averaging	8	7	700–1500	0.94	2.8	4.2	0.88
14x10	Straight	Averaging	12	4	750–1300	0.92	0.8	1.1	0.86
12x12	Duct	Averaging	16	9	700–1600	0.89	1.1	2.3	0.86
24x8		Spot & Avg	16	7	700–1400	0.93	1.5	2.2	0.87
24x24	8' long	Averaging	64	1	750	0.94	..	..	0.87
			EFFECT OF LOW VELOCITY—SEE TABLE 2.						
			EFFECT OF LENGTH OF APPROACH (*Free Openings*)						
	1″ long	Avg	16	4	800–1600	0.67	1.1	1.5	0.65
	6″ long	Avg	16	4	700–1400	0.81	1.2	1.2	0.76
12x12	12″ long	Avg	16	3	800–1300	0.87	1.1	1.1	0.82
Straight	18″ long	Avg	16	3	860–1600	0.92	1.4	2.2	0.86
Open	24″ long	Avg	16	9	700–1600	0.89	1.1	2.3	0.86
Duct	42″ long	Avg	16	3	800–1500	0.93	1.1	2.1	0.87
	52″ long	Avg	16	4	750–1500	0.87	1.5	3.5	0.85
	67″ long	Avg	16	4	800–1600	0.86	0.9	2.3	0.85
			EFFECT OF GRILLES						
20″x8″ Sq. Punched 71% Free Area		Spot	10	5	700–1800	0.88	2.1	3.5	0.86
24″x8″ V-Bar 76% F. A.		Spot	16	4	700–1600	1.01	2.5	5.0	1.00
24″x8″ H-Bar 85% F. A.		Spot	16	4	700–1500	1.01	1.0	2.0	0.97

(*Approach to Grille: Nominal Size Rectangular Duct—2 ft long.*)

center reading gave results within 2 per cent for 11 out of 14 openings. Momentary fluctuations and poor distribution will affect this factor however, and it is also in error by as much as 15 per cent for certain types of grilles. More work should be done to investigate thoroughly the advantages and the limitations of the valuable instantaneous-type instruments such as the velometer and the hot-wire devices.

Fig. 8. Typical Curves of Application Factor *vs* Air Velocity for Air Discharge Openings

A—24 in. sq open duct
B—14 in. x 8 in. open duct
C—12 in. sq open duct—1 in. long
D—12 in. sq open duct—18 in. long
E—24 in. x 8 in. open duct
F—24 in. x 8 in. low resistance grille
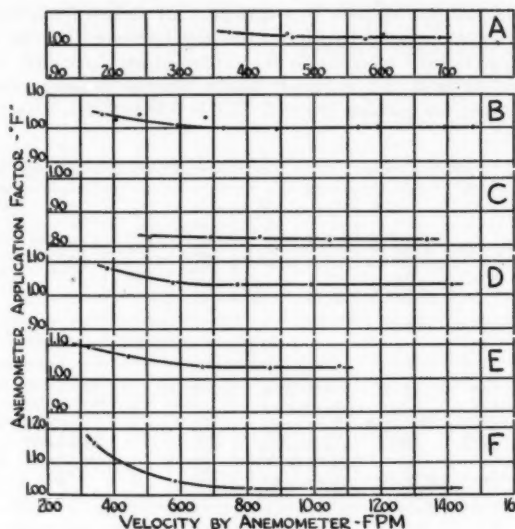
Correlations and Comparisons

The Guide 1939 recommended 0.873 for the anemometer application factor for air intakes or room exhaust grilles for all usual velocities. This was based on the work of Greene and Dean.[4] The Guide 1940 recommends a factor of 0.80 for average conditions. This was based on the earlier work of Davies.
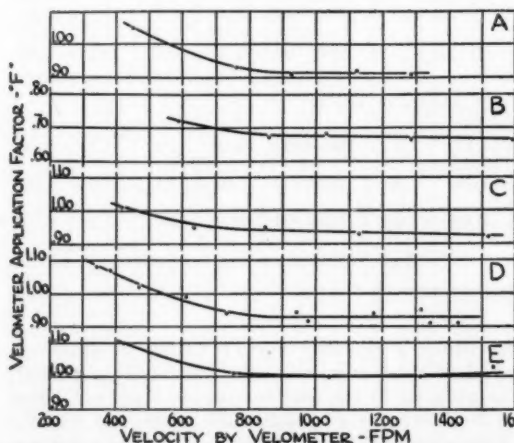
Fig. 9. Typical Curves of Velometer Application Factor *vs* Air Velocity for Air Discharge Openings

A—14 in. x 10 in. open duct
B—12 in. x 12 in. open duct —1 in. long
C—12 in. x 12 in. open duct —3½ ft long
D—24 in. x 8 in. open duct
E—24 in. x 8 in. low resistance grille

[4] ASHVE Research Report No. 1092—The Flow of Air Through Exhaust Grilles, by A. M. Greene Jr. and M. H. Dean. (ASHVE Transactions, Vol. 44, 1938, p. 387.)

A re-analysis of the data given in these two papers gives the results presented in Table 6, covering air intake openings. Davies' work included very low velocities and some grilles of very high resistance. Averaging all the original data given in the Davies papers which apply to velocities above 300 fpm and to grilles of more than 50 per cent free area, the anemometer factor is 0.836. The difference between this value and the 0.85 obtained in the present investigation is probably due mainly to the narrow flange used by Davies around the grilles he tested. Part of the difference may be due to anemometer calibrations, as Davies reported 6 per cent difference between two of the anemometers he tested. Greene and Dean show about 67 plotted points in their Fig. 17, and they give an average anemometer factor of 0.873. Their special traverse method gave the middle zone of the grille a value four times that of the top and bottom zones. Since the anemometer actually reads *highest* in the

TABLE 6. Comparison of Anemometer Factors from Various Sources

Reported by	Approx. No. of Traverses	No. of Sizes of Openings	No. of Grilles	Velocity Range	Average Factor	Average Deviation Per Cent
Air Intakes or Room Exhaust, Wall or Flange-Type Opening						
Davies.........	20	1	6	330–700	0.836	2
Greene and Dean.	67	4	20	230–1000	0.873	3
Tuve and Wright.	306	12	5	400–1500	0.85	2
Air Discharge or Room Supply—Long Duct Approach						
Davies.........	17	2	5	300–750	0.99	1.5
Larson, Nelson and Kubasta..	4	1	1	300–900	1.00	0.3
Tuve and Wright.	212	8	4	450–1500	1.03	1.5

corners, as shown by the test data of Fig. 10, any method which slights these corner values would result in an anemometer factor nearer unity. From these evidences it is concluded that for an equally-timed traverse the new factor of 0.85 is substantiated by previous work and that it is well proven by tests on at least 30 different grilles, for the velocity range above 300 fpm. For velocities below 300 fpm there is a sharp contradiction in the findings of the three investigations, and further tests should be made.

For air discharge or supply openings, THE GUIDE 1940 recommends an anemometer factor of 1.00 for velocities above 600 fpm and 0.97 for the range 150-600 fpm, using the arithmetical mean of the core area and the net free area as the multiplier to obtain volume. These were the Davies results, and they were based on a long duct approach. A re-analysis of the data at present available on air outlets in the range of velocities above 300 fpm is given in Table 6. While the data on grilles are not entirely adequate, the anemometer coefficient for a free opening is evidently above unity. There is again a complete disagreement regarding the factors to be applied at velocities below 300 fpm.

A calibration factor of unity has also been reported for an outlet grille mounted on an elbow or stack head,[5] although this factor probably depends on the character of the stack-head installation.[6] A series of additional tests on stack heads is being conducted under the direction of the ASHVE Technical Advisory Committee on Air Distribution and Air Friction at the University of Wisconsin by Prof. D. W. Nelson and will no doubt furnish ample data on such outlets.

Further Investigation

It would be profitable to extend the study of this subject to obtain more data in the range of velocities below 300 or 400 fpm, and also to test a few more arrangements of supply grilles. Since the cooperative projects at the Universities of Wisconsin and Illinois deal with air flow through stack heads and room supply outlets, further data will be available on this subject very soon.

The next major step in the study at Case School of Applied Science will be to extend the measurements into the room space, investigating both the primary and the induced air motion. Tests already made have shown that for the same size of opening and the same volume of primary air, a stream issuing from a square-edged plenum opening will have a longer throw than a stream from a straight duct, and that there is also considerable difference in the quantity of air entrained in the two cases.

The velometer and the hot-wire anemometer are especially well suited for these studies of free air streams, and supplementary data will be collected on these valuable instruments. The subject of grille performance will be studied, and the entire investigation will ultimately focus on the measurements of air velocities and drafts within the occupied zone of the room.

1145	1080	1080	1162
992	1000	1000	1105
1145	1125	1093	1150

WIDE FLANGE

Fig. 10. Typical Distribution of Anemometer Velocity Readings at an Air Intake

Velocities as read by anemometer in a traverse of a 14 in. x 10 in. flanged air intake with a 4-in. anemometer. (Total 36 readings, 2 min each.)

DISCUSSION

D. W. Nelson (Written) : It would be well at the beginning to describe the type of instruments that the paper covers. It is clear that by anemometer is meant the propeller or rotating vane type, but all air meters are anemometers by the dictionary and common usage. Later on it is described as rotating vane type and again as rotating vane wheel type, but the designation would be clearer at the beginning. I believe The Guide refers to the same type as the propeller type.

The use of trade names in place of descriptive titles such as deflecting vane anemometer as used in The Guide is objectionable. The distinction is the same as between Kodak and camera, Frigidaire and refrigerator. A specific reference to the

[5] ASHVE Research Report No. 936—Investigation of Air Outlets in Classroom Ventilation, by G. L. Larson, D. W. Nelson and R. W. Kubasta. (ASHVE Transactions, Vol. 38, 1932, p. 463.)
[6] Investigation of Warm-Air Furnaces and Heating Systems, by A. C. Willard, A. P. Kratz and V. S. Day. (University of Illinois, Engineering Experiment Station Bulletin No. 120, p. 97).

calibration of these instruments in the first paper might be worth while. I believe the deflecting vane type calibration is only given at low velocities yet in the present paper was used at high velocities with a jet attachment.

For the sake of uniformity the anemometer should face in direction of calibration; however, in calibrating a 4 in. propeller type no difference with reversal of air flow through the instrument was found.

On many grilles with thin dividing members good results are secured on the deflecting vane type with the exploring jet held 1 in. away from the grille surface and the use of the core area. The theory in this being that the individual jets have united at this distance and yet the issuing stream has not spread beyond the core area. Of course, this method would not be universally applicable to all grille types.

Agreement between the wind tunnel and whirling arm methods is given as 2 per cent. In the previous paper [7] it was given as 3 per cent. Is this the same calibration or a later one? We have not been able to get that close an agreement except by the use of a coefficient applied to the entrance cone or nozzle that varies with velocity. The nozzle is very smooth and of accepted form and the velocity across the 12 in. transparent test section is uniform except within a fractional part of 1 in. from the surface.

The propeller type anemometer, although held against the grille, has its working element some distance away, perhaps a distance of $\frac{1}{2}$ to $\frac{3}{4}$ in. On some grilles the individual jets would have perhaps reunited at that distance so a coefficient near unity might be obtained by using the core area. A disturbing factor in this would be the tendency of this type to read above the average velocity when the wheel is exposed to non-uniform velocities. That is, if one side is exposed to a velocity 50 fpm higher than the other side, the reading will be more than 25 fpm above the lower reading.

We have not been able to calibrate a deflecting vane anemometer in a wind tunnel as mentioned in this paper, except in an indirect manner. Was it actually calibrated in the jet issuing from a nozzle discharging to the atmosphere? The impression is given that orifices may be quite easily used as primary measuring devices. It perhaps should be made clearer that they must be calibrated against some other device for the conditions of use as to duct size, velocity, sharpness of edge, and thickness of plate. The complications are enough and the coefficients so variable that use on experimental work is discouraged in preference to the rounded approach nozzle. They are very convenient in sets in handling a great range of volumes and we have used them on infiltration tests having calibrated them against gas meter flow measurements.

The manufacturer of the deflecting vane instrument used describes the reversal of the exploring jet for intake opening use. Was this method tried? We have considered it practical. Also the double tube furnished with this instrument for exploring duct air streams might be used for this purpose when made with the proper range. Another method is to use the double pitot tube with sensitive pressure gages when the tube is made of very small tubing soldered together with opposed openings. This latter must be calibrated (whereas the usual pitot tube is inherently correct) and is probably not suited to as low velocities as encountered on air conditioning exhaust grilles.

It is stated that some types of anemometers were not tried because first cost was high. Yet several can be made or purchased at about the cost of the deflecting vane type. Also use of the Kata thermometer is disparaged due to unfortunate experience. We have found them very consistent and valuable although slow in use. The use of several Kata thermometers kept in a hot water bath under temperature control obviates some of this inconvenience. These instruments can be calibrated and are very excellent instruments especially for free air movement in comfort studies.

[7] Loc. Cit. Note 2.

As mentioned in the paper the approach conditions influence the instrument factor materially and more often than not the approach is poor. However, for field use and even for laboratory work acceptable accuracy can be secured. Discharge at an angle and reversal of flow, such as occurs from an expanding stack head, probably cause the greatest difficulty.

In describing the factors of near unity and 0.93 for the two types of instruments it would be well to mention that these are based on different areas. Less difference would exist if the areas were the same. In Fig. 7 it is noted that a duct approach of 1.5 diameters is required for uniform approach. In a previous study[a] a 3 ft length was found very satisfactory for grilles of various dimensional ratios and all ordinary areas.

In the stack head testing program[b] we have found it difficult to get a factor for the deflecting vane anemometer and the same would apply for most indicating instruments on widely variable flow. It is difficult for the observer to judge the middle point of the needle swing and more difficult to weight the swing with respect to time so as to get an integrated determination. We find the observer, as he learns from experience and knowledge of correlation of the true volume by nozzle measurement with the observed stack head volume, applies a mental correction factor. This complication does not exist with the propeller type which totals the reading on the dial.

The agreement of the Davies factor of 0.80 corrected to 0.836 for velocities above 300 fpm and free area grilles above 50 per cent with the present factor of 0.85 seems to be complete. Closer agreement between investigators could hardly be expected with conditions not being exactly duplicated. The lack of agreement at lower velocities could well be due to calibrating difficulties in this type of instrument which has considerable friction drag at low velocities that may vary from time to time.

In the concluding sentence of the paper *drafts* is used to describe low velocity turbulent air motion whereas by definition it is accepted as being associated with comfort due to a certain low temperature or a temperature difference associated with such low velocities.

The present paper represents a very large amount of work and brings us further towards accuracy in the use of velocity air measurements. I remember vividly when a prominent member of the Society some 10 years ago stated from the floor that he would as soon accept the results from wetting the back of the hand as from a propeller type anemometer. Papers like this one do much to instruct in the scientific method of approach.

D. J. Stewart (Written): It is troublesome in the field to arrive at the free open area of a grille, and, therefore, it is difficult to determine the designated area. No doubt the application factor varies too much to permit the use of core area with a single application factor, but if a factor could be obtained which could be used with core areas even if it were applicable only to a particular type of grille, it would be very useful for routine work in the field.

I think it would be well to have a published application factor for the movable vane type instrument for use at air intakes (room exhaust) as in many instances this is the only type available. No doubt it is not as satisfactory as the anemometer for this purpose but we have used it with results consistent enough for most practical purposes.

While the calibration of some Kata thermometers is not good and they are not by any means the best instruments for this type of work, nevertheless it is possible to use them for air discharge openings if no other instrument is available and an application factor would be useful.

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[a] ASHVE Research Report No. 1076—Air Distribution from Side Wall Outlets, by D. W. Nelson and D. J. Stewart. (ASHVE Transactions, Vol. 44, 1938, p. 77.)
[b] ASHVE Research Report No. 1156—The Performance of Stack Heads, by D. W. Nelson, D. H. Kranz and A. F. Tuthill. (ASHVE Transactions, Vol. 46, 1940, p. 205.)

EFFECT OF ROOM DIMENSIONS ON THE PERFORMANCE OF DIRECT RADIATORS AND CONVECTORS

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THE results of previous investigations [1, 2, 3, 4, 5] on the performance of direct steam radiators and convectors in test rooms 9 ft x 11 ft with 9 ft ceilings in many cases showed rather large differences between the temperature of the air near the ceiling and that near the floor. The magnitudes of these differences led to some question as to whether the results obtained in the comparatively small rooms were entirely characteristic of the performance of certain types of radiators and convectors, and whether somewhat different results might be obtained in rooms more nearly conforming in size with rooms commonly in use. The subsequent rebuilding of the room heating testing plant, and the installation of a test room 15 ft x 18 ft with an 8 ft-6 in. ceiling, afforded the opportunity to study the effect of the size of the room on the temperature gradients from floor to ceiling produced by direct steam radiators and convectors, and to determine whether conclusions with respect to the comparative performance of the different types of units were in any way affected by the size of the room.

Fig. 1. Plan Section of

Description of Apparatus

A complete description of the room heating testing plant involving two small test rooms 9 ft x 11 ft x 9 ft has been given in previous publications.[6] The test rooms were of standard frame construction, each room presenting two

[6] Loc. cit. Notes 1-5.
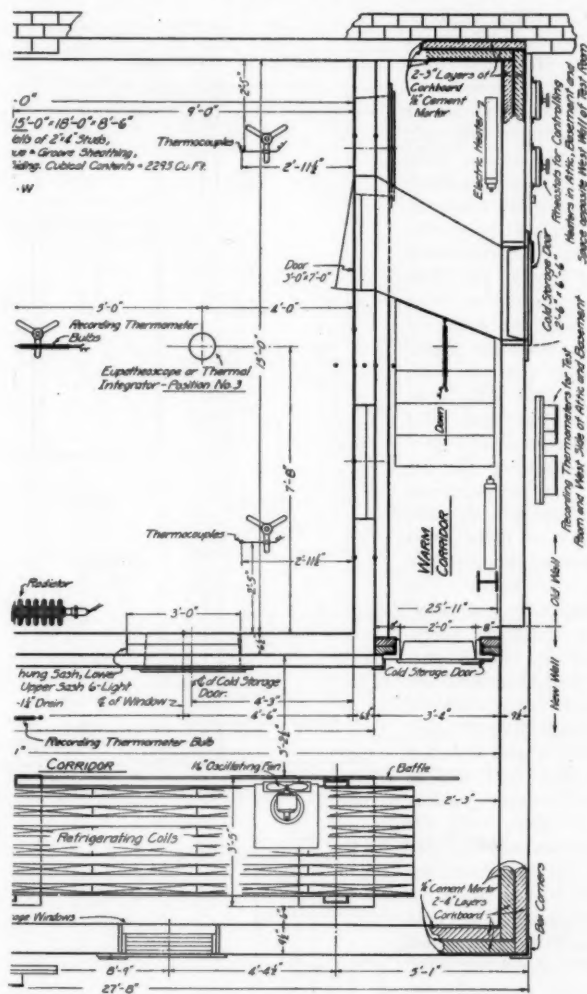
ROOM HEATING TESTING PLANT

exposed walls. Means were provided for obtaining any desired temperatures below the floors and above the ceilings. In many respects the remodeled room heating testing plant, shown in Figs. 1 and 2, was similar to the original plant, with the exception that it contained only one test room which was 15 ft wide, 18 ft long and had an 8 ft-6 in. ceiling. The test room was completely enclosed by a larger structure having walls, floor, and ceiling of 6-in. and 8-in.

cork, which formed corridors on three sides of the room and included spaces corresponding to an attic and basement. As shown in Fig. 2, the walls enclosing the attic and basement spaces were insulated and equipped with cold storage doors, and the spaces were provided with electric heaters, affording means of controlling the temperatures above the ceiling and below the floor of the test room. Refrigerating coils, shown in Fig. 1, were placed in the north corridor and shielded from the north wall of the test room by means of a baffle or radiation shield. The west corridor was equipped with cold storage doors so that it could be isolated from the other corridors. This corridor was also provided with electric heaters for controlling the air temperature, and the arrangement made it possible to operate the test room with either two or three walls exposed to any desired temperature maintained in the cold room. The walls of the test room were constructed of bevel siding, building paper, sheathing, 3⅝ in. studs, and plasterboard. The north wall contained two 3 ft x 4 ft-6 in. double-hung windows, and the east and west walls each contained a similar window and a 3 ft x 7 ft paneled wood door with a 27½ in. x 36 in. glass in the upper portion. For the purpose of these tests the radiators and convectors were located at the north wall 2½ in. from the plaster and centrally in the space between the two windows. The cold storage doors at the end of the west corridor were closed and the test room was operated with only two walls exposed. Air movement over the north wall was obtained by means of fans.

The radiators and convectors were connected with a single pipe for steam and return condensation. The condensation was collected in a receiver located in the basement of the laboratory below the test plant, and was weighed
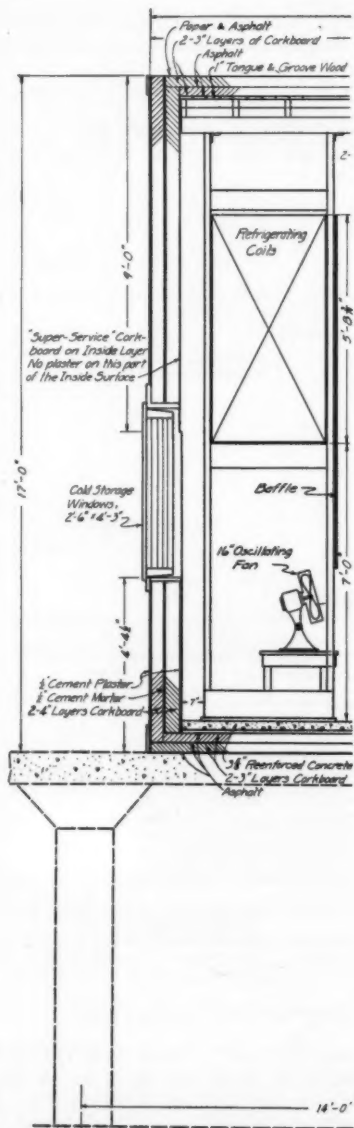
Fig. 2. Elevation
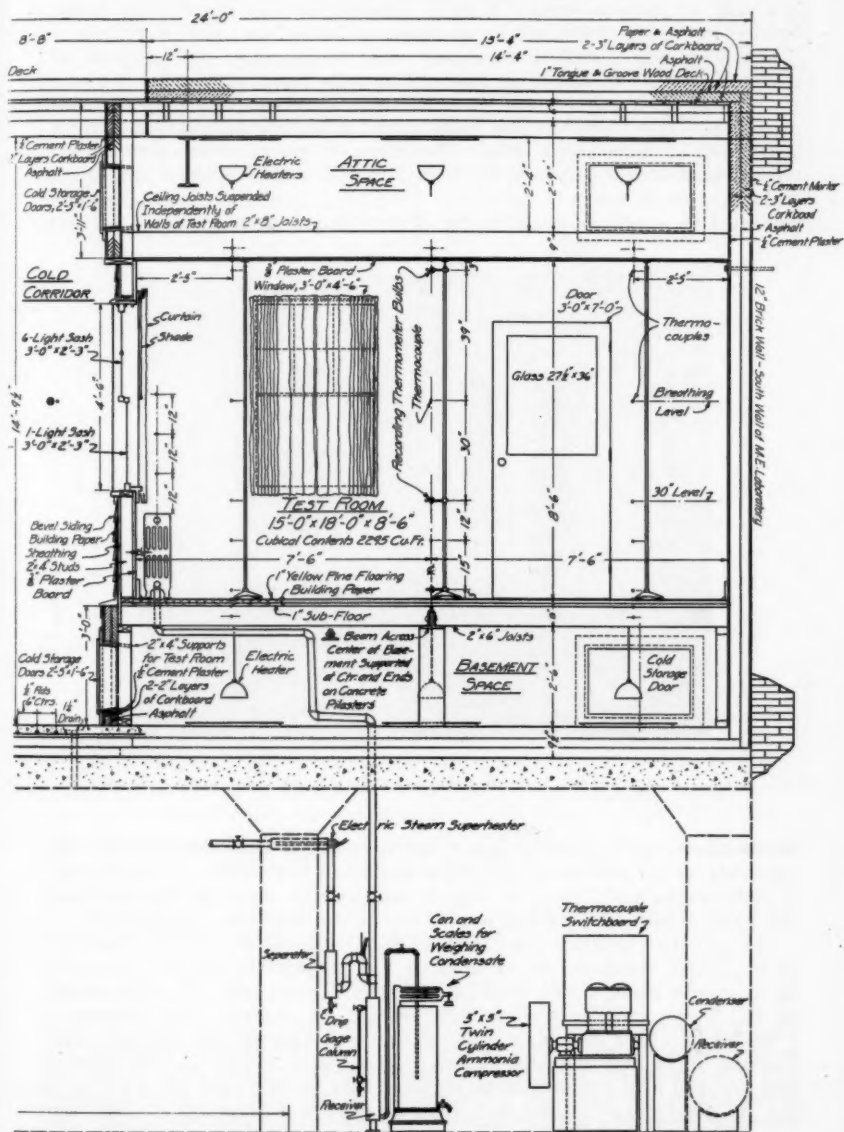
SECTION OF ROOM HEATING TESTING PLANT

in a tank on platform scales. A steam separator and an electric superheater installed in the line to the radiator or convector insured the use of dry steam.

The plant was completely equipped with thermocouples and temperature recorders in order to permit all observations to be made without the necessity for entering the test room and thus disturbing conditions at any time during the test period. Thermocouples were located at various points on the inside and outside surfaces of the walls. Thermocouples for observing air temperatures were located 3 in. above the floor, 3 in. below the ceiling, at the 18 in., 30 in., and 60 in. levels in the room. Five sets of these thermocouples were supported on standards placed in the five positions shown in Fig. 1. These
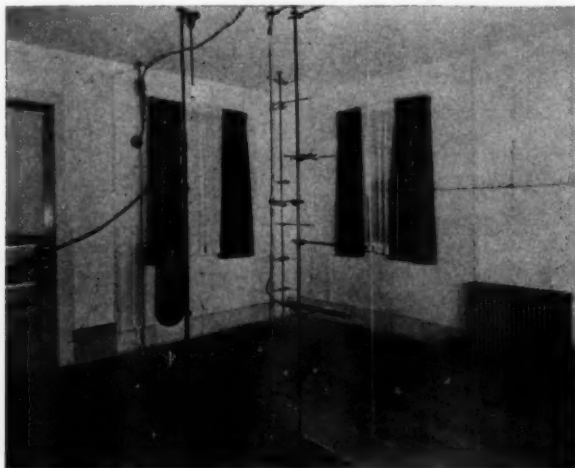
FIG. 3. THE THERMO-INTEGRATOR INSIDE OF TEST ROOM

thermocouples were connected into a switch system so that they could be read separately, or the average of the five at any given level could be read directly. A thermo-integrator,[7] shown in Fig. 3, and used for observing the equivalent and mean radiant temperatures was located in Position 3, as shown in Fig. 1.

The cast-iron radiators used in these tests consisted of: (1) a 13-section, 26 in., 5-tube radiator, with sections on $2\frac{1}{2}$ in. centers; (2) an 18-section, 26 in., 3-tube radiator with sections on $2\frac{1}{2}$ in. centers; and (3) a 25-section, 25 in., 3-tube radiator with sections on $1\frac{3}{4}$ in. centers. The convectors consisted of (1) a $5\frac{3}{8}$ in. x $16\frac{1}{8}$ in. cast-iron heating unit equipped with a cabinet 26 in. high and $69\frac{1}{8}$ in. long; (2) a $5\frac{5}{8}$ in. x 6 in. cast-iron heating unit equipped with a cabinet 26 in. high and $56\frac{7}{8}$ in. long; and (3) a $5\frac{1}{2}$ in. x $2\frac{1}{2}$ in. non-ferrous heating unit equipped with a cabinet 26 in. high and

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[7] The Thermo-Integrator—A New Instrument for the Observation of Thermal Interchanges, by C-E. A. Winslow and Leonard Greenburg. (ASHVE TRANSACTIONS, Vol. 41, 1935, p. 149.)

Cold Room Temp. in deg. F.

	Temperature in deg. F.					
	30" Level Center of Room	Integrator at Integrator	Integrator in Position No.3 Equivalent	Mean Radiant	Difference Between Air at Integrator and Equiv	M. Rad
-1.4 9.85 lb. Test R-32, 25, 3-Tube Radiator	67.9	67.5	67.5	67.5	0.0	0.0
-1.3 10.11 lb. Test R-31, 25, 3-Tube Radiator	68.9	68.6	68.4	68.4	0.2	0.2
-1.0 9.61 lb. Test R-26, 26, 3-Tube Radiator	68.2	67.9	67.6	67.5	0.3	0.4
-1.2 10.02 lb. Test R-22, 26, 5-Tube Radiator	68.2	68.0	67.8	67.7	0.2	0.3
-3.1 10.69 lb. Test R-20, 26, 5-Tube Radiator	68.8	68.5	68.4	68.4	0.1	0.1

0 2 4 6 8 10 12
Net lb. of Steam Condensed per hr.

13 Sec., 26", 5-Tube Radiator
Temp. at Ctr. of Test Room
Test R-20, No Wrapping
Test R-22, 2 Sec. Wrapped
Max. Temp. Min. Temp. at Given Level

13-Section, 26 In., 5-Tube
Cast Iron Radiator
Sections on 2½ In. Ctrs.
Length = 32½ In.

18 Sec., 26", 3-Tube Radiator
Temp. at Ctr. of Test Room
Test R-26, No Wrapping
Max. Temp. Min. Temp. at Given Level

18-Section, 26 In., 3-Tube
Cast Iron Radiator
Sections on 2½ In. Ctrs.
Length = 45 In.

25 Sec., 25", 3-Tube Radiator
Temp. at Ctr. of Test Room
Test R-31, No Wrapping
Test R-32, 1½ Sec. Wrapped
Max. Temp. Min. Temp. at Given Level

25-Section, 25 In., 3-Tube
Cast Iron Radiator
Sections on 1½ In. Ctrs.
Length = 43½ In.

Air Temperature in deg. F.

Floor 30" Level Breathing Level Ceiling

-1 0 1 2 3 4 5 6 7 8 9 10
Height Above Floor in Feet

FIG 4. PERFORMANCE CURVES FOR THREE TYPES OF DIRECT CAST-IRON RADIATORS IN 15 FT X 18 FT X 8 FT-6 IN. TEST ROOM

50¹/₁₆ in. long. Details of all the radiators and convectors used in both test rooms are shown on the different curve sheets.

TEST PROCEDURE

In all cases the test room was operated with two walls exposed and the temperature in the cold room was maintained at from −1.0 to −3.0 F, with an equivalent wind velocity of approximately 10 mph over the north wall. The temperature of the air above the ceiling of the test room was maintained at 62 F, and the temperature of the upper surface of the floor was maintained from 1.0 to 1.5 F higher than that of the lower surface. All tests were run with two walls exposed and the inside surface of the wall between the test

Cold Room Temp. in deg. F.

Net lb of Steam Condensed per hr.

		Temperature in deg. F.					
		30° Level		Integrator in Position No 3		Difference Between Air at Integrator and	
Cold Room Temp. in deg. F.		Center of Room	at Integrator	Equivalent	Mean Radiant	Equiv.	M.Rad.
-1.2	9.33 lb. Test C-17 Convector No.44	68.1	69.1	67.4	66.6	1.7	2.5
-0.3	10.93 lb. Test C-16 Convector No.44	74.2	75.0	72.7	71.6	2.3	3.4
-1.0	9.22 lb. Test C-4 Convector No.43	68.2	68.9	67.1	66.2	1.8	2.7
-1.1	9.91 lb. Test C-2 Convector No.43	70.3	71.1	69.0	67.9	2.1	3.2
-1.6	9.29 lb. Test C-9 Convector No.42	68.2	68.8	67.2	66.4	1.6	2.4
-1.7	10.97 lb Test C-8 Convector No.42	74.2	74.7	72.3	71.1	2.4	3.6

0 2 4 6 8 10 12

Convector No. 42
Temp. at Ctr. of Test Room
Test C-8, No Blocking
Test C-9, 22" Blocked
Max.Temp. / Min.Temp. at Given Level

Convector No. 42
Cast Iron Heating Unit
Cabinet Length = 69½ In.

Convector No. 43
Temp. at Ctr. of Test Room
Test C-2, No Blocking
Test C-4, 11" Blocked
Max.Temp. / Min.Temp. at Given Level

Convector No. 43
Cast Iron Heating Unit
Cabinet Length = 56⅞ In.

Convector No. 44
Temp. at Ctr. of Test Room
Test C-16, No Blocking
Test C-17, 12" Blocked
Max.Temp. / Min.Temp. at Given Level

Convector No. 44
Non-ferrous Heating Unit
Cabinet Length = 50¼ In.

Air Temperature in deg. F.

Floor — 30 Level — Breathing Level — Ceiling

Height Above Floor in Feet

Fig. 5.   Performance Curves for Three Types of Convectors in 15 ft x 18 ft x 8 ft-6 in. Test Room

room and the closed west corridor was maintained at a temperature from 1.0 to 1.5 F higher than that of the outside surface. No test observations were made until conditions had remained constant for several hours, as indicated by the readings of the air temperature and that on the inside wall surfaces. A saturation temperature of 215 F was maintained for the steam in the test units, and when thermal constancy had been attained, the condensate was weighed over the period of one hour. Weighings were made at intervals of 10 min. The condensate was corrected by subtracting the condensation occurring in the piping alone, which was determined by separate tests.

RESULTS OF TESTS

Performance in Large Test Room

The steam condensations and the temperature gradients produced in the room by the three radiators tested are shown in Fig. 4. Similar curves for the convectors are shown in Fig. 5. Details of both the radiators and the convectors are shown in the accompanying insets. For the purpose of comparison all of the units tested were operated to maintain a temperature of 68 F at the 30 in. level in the center of the room. Since it was not always possible to select commercial units of exactly the size required to maintain 68 F at the 30 in. level, units of somewhat larger size were installed, and in the case of radiators the desired temperature was obtained by wrapping one or more sections with cloth. In the case of convectors the desired result was obtained by blocking part of the heating unit and cabinet.

Fig. 4 shows that with the 26 in., 3-tube radiator, 18 sections were just sufficient to give a temperature of 68 F at the 30 in. level in the center of the room. The curves for the 26 in., 5-tube and the 25 in., 3-tube radiators indicate that in each case the effect of wrapping was to lower the temperature in the center of the room equally at all levels, and to decrease the steam condensation. That is, the new temperature gradients with some of the heating surfaces wrapped were parallel to the original ones, indicating that the wrapping had no disturbing effect, and that the wrapped radiators performed in essentially the same manner as unwrapped radiators of the correct size. The curves in Fig. 5 exhibited the same characteristics, thus indicating that a convector with a portion blocked also performed in essentially the same manner as an unblocked convector of the correct size.

In both Fig. 4 and Fig. 5 the minimum and maximum temperatures observed at the different levels in any of the positions of the thermocouples located on the 5 standards shown in Fig. 1 have been plotted, together with the temperatures observed in the vertical central axis of the room. The latter temperatures are represented by the heavy lines. The shaded areas represent the locus of the maximum deviations at any level from the temperature observed at the center of the room. It may be observed that in practically all cases the temperatures observed in the central axis represented the highest temperatures in the room. The maximum deviations in most cases occurred in the level nearest the floor and were of the order of 3½ F. The mean deviation was of the order of 2 F. Comparison of similar plots for wrapped and unwrapped radiators and blocked and unblocked convectors proved that the deviations were practically of the same magnitude and character whether the test units were of the correct size or whether the heat output was reduced by wrapping or blocking. This confirmed the conclusion that wrapping or blocking did not disturb either the temperature distribution or the temperature gradients produced in the room by a given type of test unit, and that comparisons made with the wrapped or blocked units were just as valid as those made with radiators or convectors of the correct size to maintain 68 F at the 30 in. level, as long as such comparisons did not involve the heat output per square foot of superficial area. The curves furthermore indicate that the temperature gradients observed in the central axis of the room were characteristic of those in other parts of the room, and could be validly used for comparisons of the performance of the different types of test units.
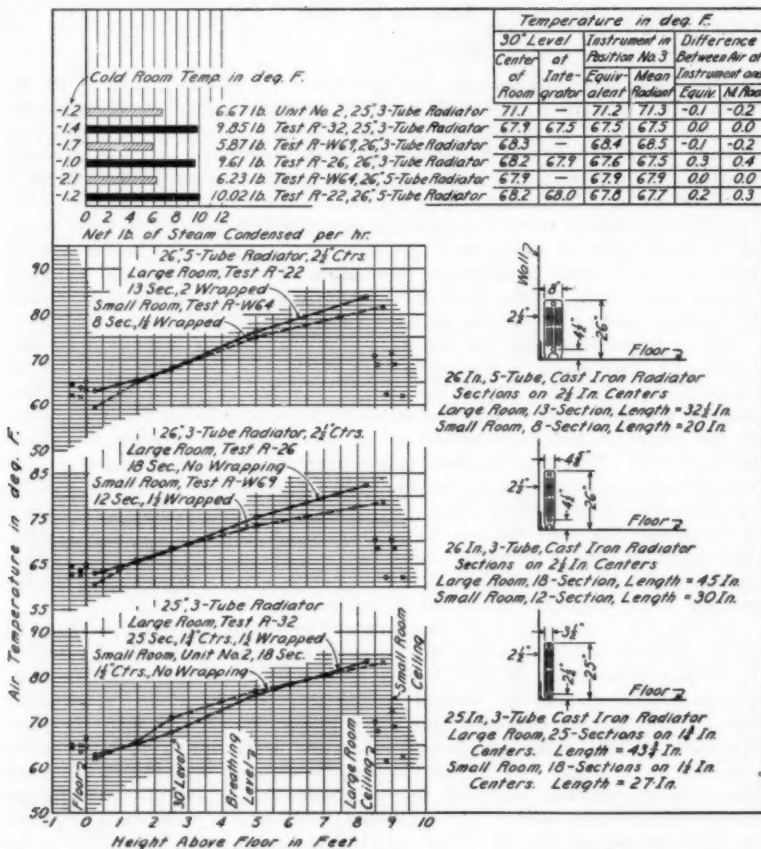
Fig. 6. Performance Curves for Three Types of Direct Cast-iron Radiators in 15 ft x 18 ft x 8 ft-6 in. and 9 ft x 11 ft x 9 ft Test Rooms

Effect of Size of Test Room

Comparisons between the temperature gradients produced in the small and large test rooms by radiators of three different types are shown in Fig. 6, and similar comparisons for three different types of convectors are shown in Fig. 7. In general it may be observed that the temperature near the floor in the large test room was approximately 4 F higher than that in the small room, while the temperature near the ceiling was from 1.0 to 3.0 F higher than that in the small room. With one exception this resulted in somewhat smaller temperature differences between the ceiling and floor in the large test room, as shown in Table 1. The last column of this table indicates that

TABLE 1—TEMPERATURE DIFFERENCE BETWEEN AIR AT CEILING AND FLOOR

DESCRIPTION OF RADIATORS AND CONVECTORS		TEST NO.	TEMP. DIFF. BETWEEN CEILING AND FLOOR AT CENTER OF ROOM IN DEG F		DECREASE IN TEMP. DIFF. OBTAINED IN LARGE ROOM DEG F
			Small Room	Large Room	
Cast-Iron Radiator	26-in., 5-tube...........	R-W64	22.1	...	1.5
	26-in., 5-tube.........	R-22	...	20.6	...
Cast-Iron Radiator	26-in., 3-tube.........	R-W69	18.2	...	−1.2
	26-in., 3-tube.........	R-26	...	19.4	...
Cast-Iron Radiator	25-in., 3-tube.........	Unit No. 2	21.1	...	0.6
	25-in., 3-tube.........	R-32	...	20.5	...
Convector	No. 1, 5⅜ in. x 14¾₆ in.	I-W101	20.8	...	2.4
Cast-Iron Heating Unit	No. 42, 5⅝ in.x16⅛ in.	C-9	...	18.4	...
Convector	No. 32, 5⅝ in. x 6 in..	Unit No. 3	23.9	...	5.0
Cast-Iron Heating Unit	No. 43, 5⅝ in. x 6 in..	C-4	...	18.9	...
Convector	No. 6, 5⅜ in. x 2 in....	I-W74	24.1	...	4.7
Non-Ferrous Heating Unit	No. 44, 5½ in. x 2½ in.	C-17	...	19.4	...

the effect of the large room was more marked in the case of convectors than it was in the case of direct radiators.

The higher temperature near the floor of the large test room was probably brought about by the fact that any air inleakage occurring at the door was less effective at the center of the large room than it was at the center of the small room. In general there was a tendency for the temperature near the ceiling of the large test room to be equal to or greater than that for the small room. In any event the magnitudes of the temperature differences between the ceiling and floor obtained with the different units in the large and the small test rooms were approximately of the same order and the deviations between the effects of the two rooms do not seem sufficient to invalidate any of the conclusions drawn in previous publications [8] from tests made in the small room.

From Figs. 6 and 7 it may be observed that in general the differences in steam condensation obtained from different units tested in the large room were considerably less than the corresponding differences obtained from the same types of units tested in the small room. Apparently, therefore, the small test room tended to accentuate any difference in performance existing between units of different types.

Effect of Type on Performance of Radiators and Convectors

In a previous paper [9] it was indicated that when radiators or convectors were operated to maintain a temperature of 68 F at the 30 in. level in the

[8] Loc. Cit. Notes 1-5.
[9] Loc. Cit. Note 3.
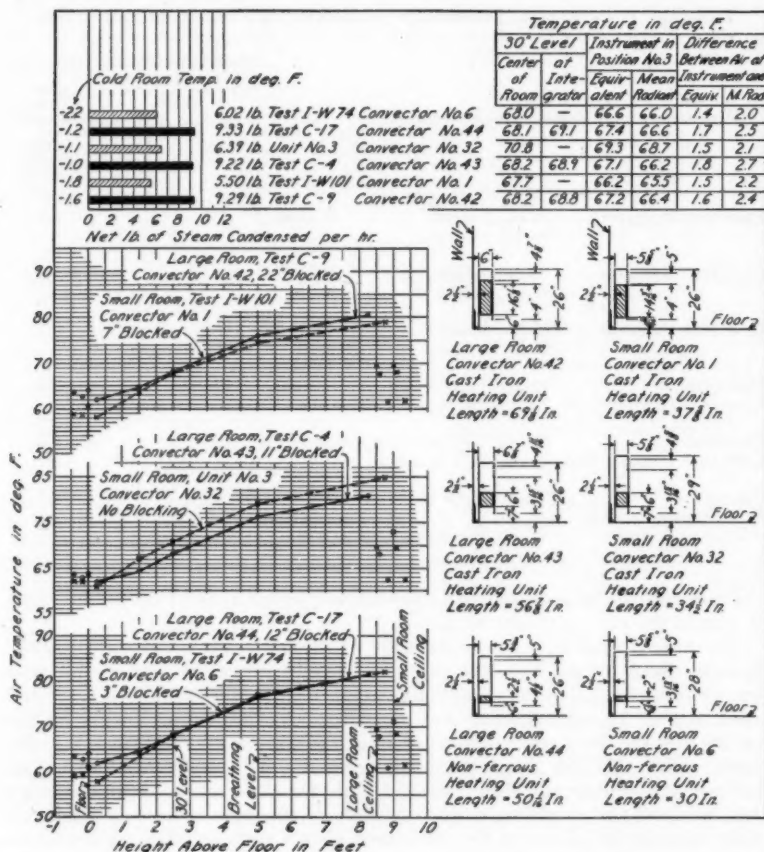
FIG. 7. PERFORMANCE CURVES FOR THREE TYPES OF CONVECTORS IN 15 FT X 18 FT X 8 FT-6 IN. AND IN 9 FT X 11 FT X 9 FT TEST ROOMS

test room, the units represented by the temperature gradient curves having the lesser slope above the 30 in. level also gave the lower steam condensations. In Fig. 8 the curves and data have been shown in groups arranged to make possible a direct comparison of the performance of the different types of radiators tested in each of the two test rooms. Similar data for the convectors are shown in Fig. 9. In the latter case data for the performance of the 25 in., 5-tube radiator have also been added for the purpose of comparison.

From Fig. 8 it may be observed that, in the small test room, the temperature gradient curves in descending order were designated as, 26 in., 5-tube and

	Temperature in deg. F.					
	30" Level Center of Room	Instrument at Integrator	Instrument in Position No.3		Difference Between Air at Instrument and	
			Equivalent	Mean Radiant	Equiv.	M.Rad.
-1.7 5.87 lb. Test R-W69, 26", 3-Tube Radiator	68.3	—	68.4	68.5	-0.1	-0.2
-2.1 6.23 lb. Test R-W64, 26", 5-Tube Radiator	67.9	—	67.9	67.9	0.0	0.0
-1.4 9.85 lb. Test R-32, 25", 3-Tube Radiator	67.9	67.5	67.5	67.5	0.0	0.0
-1.0 9.61 lb. Test R-26, 26", 3-Tube Radiator	68.2	67.9	67.6	67.5	0.3	0.4
-1.2 10.02 lb. Test R-22, 26", 5-Tube Radiator	68.2	68.0	67.8	67.7	0.2	0.3

Cold Room Temp. in deg. F.

Net lb. of Steam Condensed per hr.

Test R-W64, 8 Sec., 26", 5-Tube Radiator with 1½ Sec. Wrapped

Test R-W69 12 Sec., 26", 3-Tube Radiator 1½ Sec. Wrapped

Small Room

Large Room

Test R-22, 13 Sec., 26", 5-Tube Radiator with 2 Sec. Wrapped

Test R-32, 25 Sec., 25", 3-Tube Radiator, 1½ Sec. Wrapped

Test R-26 18 Sec., 26", 3-Tube Radiator

Air Temperature in deg. F.

Height Above Floor in Feet

26 In., 5-Tube Cast Iron Radiator Sections on 2¼ In. Ctrs. Small Room, 8-Section Length = 20 In. Large Room, 13-Section Length = 32½ In.

26 In., 3-Tube Cast Iron Radiator Sections on 2¼ In. Ctrs. Small Room, 12-Section Length = 30 In. Large Room, 18-Section Length = 45 In.

25 In., 3-Tube, Cast Iron Radiator Large Room, 25-Sections on 1½ In. Centers. Length = 43¾ In.

FIG. 8. COMPARISON OF PERFORMANCE CURVES FOR THREE TYPES OF DIRECT CAST-IRON RADIATORS

26 in., 3-tube, with steam condensations of 6.23 and 5.87 lb per hours respectively. In the large test room the temperature gradient curves in descending order were designated as 26 in., 5-tube, 25 in., 3-tube and 26 in. 3-tube, with steam condensations of 10.02, 9.85 and 9.61 lb per hour respectively. That is, the steam condensations were consistently in the order of the arrangement of the corresponding temperature gradient curves above the 30 in. level.

From Fig. 9 it may be observed that, in the small test room, the temperature gradient curves in the descending order were designated as Convector No. 6, 26 in., 5-tube radiator, and Convector No. 1 with steam condensations of 6.02, 6.23 and 5.50 lb per hour respectively. In this case there was a slight inconsistency in the location of the upper part of the curve for the radiator which was used to designate the location of the curve. However, if the lower section of the curve for the radiator were taken into consideration the inconsistency is not so marked. The curves and steam condensations for the convectors showed consistent correlation in the arrangement. In the large test room the temperature gradient curves in the descending order were designated as, 26 in., 5-tube radiator, Convector No. 44, Convector No. 43, and Convector No. 42 with corresponding steam condensations of 10.02, 9.33, 9.22
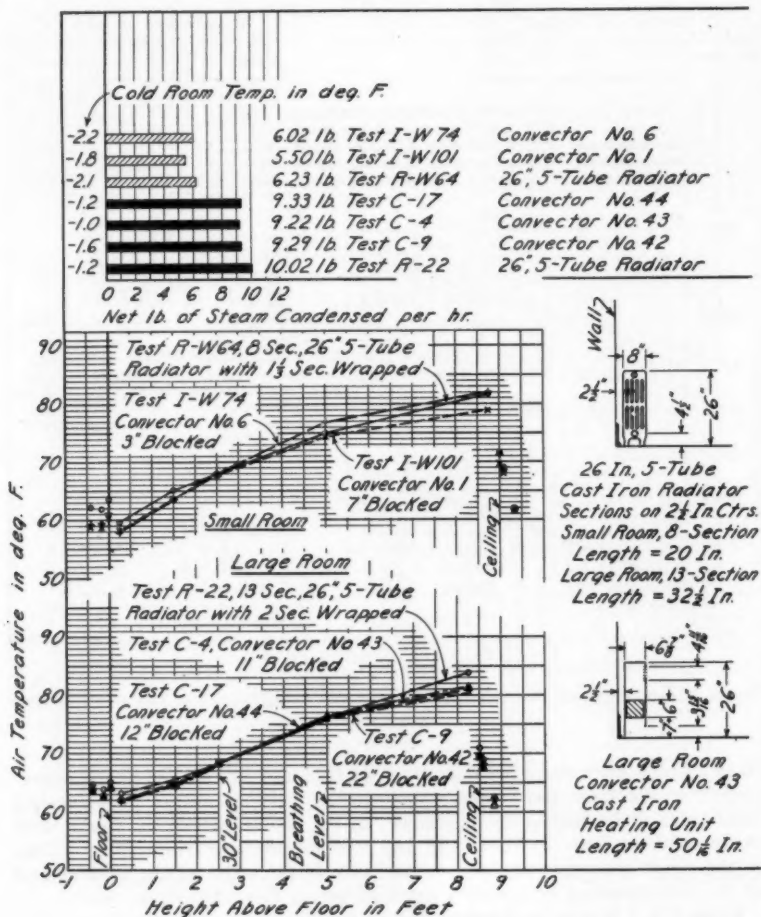
FIG. 9. COMPARISON OF PERFORMANCE

and 9.29 lb per hour respectively. Convector No. 44 was of the same general type as Convector No. 6, and Convector No. 42 was of the same type as Convector No. 1. In this case the location of temperature gradient curves was not exactly consistent with the steam condensations shown for the convectors. Since the differences under consideration were very small it is not surprising that a few inconsistencies might occur. On the whole, however, the results confirm the previous conclusion that the lesser temperature gradients are accompanied by the lower steam condensations. Furthermore while tests

| Temperature in deg. F. | | | | | |
| 30" Level | | Instrument in Position No.3 | | Difference Between Air at Instrument and | |
Center of Room	at Inte- grator	Equiv- alent	Mean Radiant	Equiv.	M. Rad.
68.0	—	66.6	66.0	1.4	2.0
67.7	—	66.2	65.5	1.5	2.2
67.9	—	67.9	67.9	0.0	0.0
68.1	69.1	67.4	66.6	1.7	2.5
68.2	68.9	67.1	66.2	1.8	2.7
68.2	68.8	67.2	66.4	1.6	2.4
68.2	68.0	67.8	67.7	0.2	0.3
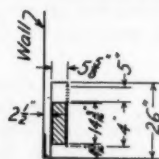
Small Room
Convector No. 1
Cast Iron
Heating Unit
Length = 37⅜ In.
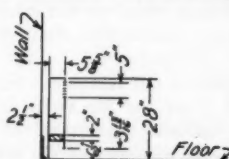
Small Room
Convector No. 6
Non-ferrous.
Heating Unit
Length = 30 In.
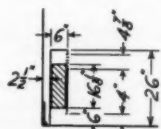
Large Room
Convector No. 42
Cast Iron
Heating Unit
Length = 69⅝ In.

Large Room
Convector No. 44
Non-ferrous
Heating Unit
Length = 50½ In.

CURVES FOR THREE TYPES OF CONVECTORS

made in the small test room tend to accentuate differences in the performance characteristics of different types or proportions of radiators and convectors, results obtained in the large test room are in the main consistent with those obtained in the small test room.

Equivalent and Mean Radiant Temperatures

The equivalent temperature of a non-uniform environment may be defined as the temperature of a uniform environment, which will result in the same heat loss from a sizable body at a given body surface temperature as that which was obtained from the same body in the non-uniform environment under consideration. A uniform environment is one in which the air and all radiating surfaces are at the same temperature. A non-uniform environment is one in which the air and all or part of the radiating surfaces are at different temperatures. The mean radiant temperature of an environment in which a body is losing heat at a given body surface temperature is an hypothetical temperature, representing the temperature of completely enclosing surfaces all at the same temperature, which would result in the same heat loss by radiation as that occurring from the given body in the given environment. The equivalent and mean radiant temperatures shown in Figs. 4 to 9 were obtained by using a thermo-integrator [10] in Position No. 3 shown in Fig. 1.

The equivalent and mean radiant temperatures shown in Figs. 8 and 9 were all obtained under the same conditions, with 68 F at the 30 in. level, and hence afford the best means for comparing the results given by direct radiators

[10] Loc. Cit. Note 7.

and convectors. The equivalent temperature, to a limited extent, correlates with the comfort chart. An equivalent temperature of 72 F with relative humidity of 25 per cent roughly correlates [11] with an effective temperature of 66 F. Equivalent temperatures lower than this probably indicate an environment too cold for comfort, and those higher an environment too warm. Data are lacking to exactly correlate these equivalent temperatures with effective temperatures below and above 66 F.

Equivalent temperatures obtained with the direct radiators, as shown in Fig. 7, were all within 0.3 F of the air temperature while those obtained with the convectors were from 1.4 F to 1.8 F lower than the air temperature. Thus, while none of the units produced an equivalent temperature that was sufficient for maximum comfort at the 30 in. level with the same temperature of 68 F at this level, the direct radiation from the radiators tended to create an environment that was slightly more comfortable than that produced by the convectors.

As shown in Fig. 8, the mean radiant temperatures obtained with the direct radiators were also all within 0.4 F of the air temperature, indicating that the direct radiation from the hot surfaces was sufficient to practically offset the chilling effect of the cold walls and windows of the test room. In the case of convectors, however, the mean radiant temperature was from 2.0 F to 2.7 F lower than the air temperature, indicating that the absence of direct radiation from hot surfaces resulted in a predominance of the chilling effect of the cold walls and windows of the room.

CONCLUSIONS

The following conclusions may be drawn from the results of these tests:

1. The use of a test room as small as 9 ft x 11 ft x 9 ft tends to accentuate differences in performance inherent in radiators and convectors of different types or proportions.

2. The use of a test room as large as 15 ft x 18 ft x 8 ft-6 in. tends to reduce differences in performance inherent in radiators and convectors of different types or proportions, but does not alter the performance characteristics sufficiently to materially affect conclusions drawn from similar tests made in a smaller test room.

3. Radiators or convectors which produce the smallest temperature gradients, or the least temperature differences between the air near the ceiling and that near the floor, while maintaining a temperature of 68 F at the 30 in. level, also tend to give the smallest steam condensations.

4. With an air temperature of 68 F at the 30 in. level, neither radiators nor convectors give equivalent temperatures sufficiently high for maximum comfort at this level. The equivalent temperature given by radiators is somewhat higher than that given by convectors.

5. In some parts of the room the direct radiation from radiators is sufficient to offset the chilling effect of cold walls, but with convectors the chilling effect of the walls and windows is a predominant factor.

ACKNOWLEDGMENT

[11] Loc. Cit. Note 5.

FALL MEETING, 1940

Houston, Tex.

IN AN autumn setting and with delightful weather members of the Society enjoyed the hospitality of Houston for the 1940 Fall Meeting held at the Rice Hotel, October 14 and 15. Members from 11 states were present and representatives from both the Atlantic and Pacific Coasts were among those registered.

For those who came early the Committee on Arrangements planned some pre-meeting entertainment, and 20 members and ladies saw the Rice *vs.* L. S. U. football game in Rice Stadium, Saturday night, October 12. On Sunday morning George Maves led the golfers to Brae Burn Country Club, and in the afternoon, W. R. Etié conducted a sightseeing group of 14 to San Jacinto Monument, the Houston Ship Channel, through Galveston, and several oil fields.

The Reception Committee was on hand early Monday morning to greet the arriving members, ladies and guests, and 152 registered for the technical sessions and nearly 200 attended the banquet.

The first session was called to order by Pres. F. E. Giesecke promptly at 10:00 a.m., Monday, October 14, at the Rice Hotel, and a·brief address of welcome was delivered by Robert J. Cummins, president of the Texas Section of *ASCE*. An appropriate response to this hearty welcome was made by Pres. F. E. Giesecke, after which he introduced J. van O. Weaver, vice-chairman of the Committee on Arrangements. He expressed the regrets of Chairman C. A. McKinney, who was called away unexpectedly and could not return in time for the meeting.

Technical discussions were resumed when President Giesecke called the second session to order at 10:00 a.m., October 15, and First Vice-President Fleisher presided. F. C. Houghten, director of the ASHVE Research Laboratory, Pittsburgh, gave an interesting talk on Our Present and Future Research Program. He gave a brief resumé of the problems being investigated and presented some details of the work on ducts and radiant heating.

Among those who asked questions were President Giesecke, College Station, Professor Kratz, Urbana, R. B. Bartlett, Bloomington, Ill. and M. W. Brown, Dallas.

E. K. Campbell, Kansas City, offered the following resolutions which were unanimously adopted.

WHEREAS, THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS assembled in Houston, Texas, for the Southwestern Regional Fall Meeting 1940, desires to express appreciation for the many courtesies received.

BE IT RESOLVED THAT the Committee on Arrangements and all of the Sub-Committees of South Texas Chapter be complimented on the arrangements which have been so well planned and so smoothly carried out for the enjoyment of everybody present.

THAT, we especially commend the efforts of the ladies of the South Texas Chapter who have made the stay of the visiting ladies so pleasant and enjoyable.

THAT, we appreciate highly the program arranged by the Banquet Committee. We liked the humor of the toastmaster, John F. Scott. We thoroughly enjoyed the able address of the President of A. & M. College, Dr. T. O. Walton.

THAT, we thank the Rice Hotel for their service, the unfailing courtesies of all of the employees and the cooperation of the management in making our meeting a success.

THAT, we appreciate the contributions of the authors of papers for their presence and the very valuable data presented, also the extremely interesting discussions.

THAT, we have greatly enjoyed the sightseeing trips to see things that are peculiar to the Gulf Coast—the Ship Channel, the San Jacinto Battleground Memorial, Galveston and its sea wall, and some of the many air conditioning installations and industrial plants in Houston.

THAT, we are greatly pleased with the cooperation rendered by the staff and the services furnished by the Houston Chamber of Commerce to assure the success of our meeting.

THAT, we enjoyed the inspiring luncheon talk of W. M. Ryan on the History of Texas.

THAT, we appreciate the unusual interest of the newspapers and their reporters, who have been attentive and generous in publicizing our meeting and interviewing our officers and many of our members.

THAT, the notable work of the Publicity Committee be recognized for the volume and extent of the coverage, which indicates careful and intelligent planning, and

FINALLY THAT, it is a great pleasure to thank the South Texas Chapter for fulfilling its promise that the finest kind of weather would prevail at this time of year; we have enjoyed the beautiful weather to the utmost, we have had a wonderful time, and we are glad we came to Houston.

As no further business was brought before the meeting it was voted to adjourn.

Entertainment

Monday at noon a get-together luncheon was served to 90 members, guests and ladies. A. J. Rummel, President of South Texas Chapter, presided and introduced the speaker, W. M. Ryan, a prominent attorney of Houston, whose subject was The History of Texas. The address was most interesting and inspiring and President Giesecke expressed the thanks of the members to Mr. Ryan at the conclusion of his talk.

During the afternoon many of the members and ladies enjoyed a sightseeing trip through Houston's fine residential section, the campus of Rice Institute, San Jacinto Battleground and nearby oil fields and refineries.

Banquet

The banquet was held at 7 o'clock in the grand ballroom and nearly 200 were present to enjoy an excellent menu featuring Texas beef steak. At the conclusion of dinner John F. Scott presided as toastmaster and President

Giesecke introduced distinguished guests, the officers of the Society, Council members and the Chapter officers and committees.

Dr. T. O. Walton, president of A. & M. College of Texas, gave an entertaining address on the Industrial Development of Texas.

After dinner the floor was cleared for dancing, which was enjoyed until 12:30, to the music of John Sullivan and his orchestra.

The program was planned by A. M. Chase, Jr., and the members of his Banquet Committee. The interesting table decorations of cotton bolls, rice heads, and pumpkins were arranged by Mrs. Chase.

At the invitation of the Houston Junior Chamber of Commerce many of the members attended a special luncheon and heard a fine address by Stanley Foran on Americanism—Let's Resell it to the World.

PROGRAM OF FALL MEETING

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

RICE HOTEL, HOUSTON, TEX. OCTOBER 14-15, 1940

October 12

1:00 P.M. Reception of Members by South Texas Chapter Committee.
8:15 P.M. Football Game, Rice Stadium (Rice vs. L.S.U.—Tickets $2.50 per person).

October 13

10:00 A.M. Golf—Brae Burn Country Club (Greens Fee $2.00 per person).
2:00 P.M. Bayshore Drive and Visit to Galveston.

October 14

8:30 A.M. Registration (Mezzanine Floor).
10:00 A.M. Technical Session—
Development of Instruments for the Study of Air Distribution in Rooms, by A. P. Kratz, A. E. Hershey and R. B. Engdahl.
Cooler Footcandles for Air Conditioning by W. G. Darley.
Chemical Dehumidification Agents, by F. R. Bichowsky.
12:15 P.M. Get-together Luncheon—($1.00 per person) *Grand Ball Room.*
1:45 P.M. Sightseeing Trip (register in advance, tickets $1.00)—City trip to Rice Institute, Battle Ground, Sylvan Beach.
1:45 P.M. Informal Golf Maches.
2:00 P.M. Council Meeting.
4:30 P.M. Friendship Hour for Members and Ladies (*Sam Houston Room*).
7:00 P.M. Informal Banquet and Dance (tickets $2.50 per person) *Grand Ball Room*—Address: Dr. T. O. Walton, President, A. and M. College, The Industrial Development of Texas.—John T. Scott, Toastmaster.

October 15

9:00 A.M. Registration.
10:00 A.M. Technical Session—
Weather Conditions in Texas *vs.* Human Comfort, by H. E. Degler.
Our Present and Future Research Program, by F. C. Houghten.
Direct Evaporative Cooling for Homes in the Southwest, by A. J. Rummel.
10:45 A.M. Ladies Bridge Luncheon and Style Show.
12:00 P.M. Junior Chamber of Commerce Luncheon—*Speaker* Stanley Foran. *Subject:* Americanism—Let's Resell It to the World.

2:00 p.m. Inspection Trips:
1. Champion Fibre & Paper Co. plant.
2. Deepwater Power Plant, Houston Lighting & Power Co.
3. Houston Ship Channel.
4. Typical Office Building Air Conditioning Installations: (a) 1000 ton reciprocating; (b) 800 ton centrifugal; (c) 350 ton absorption.

COMMITTEE ON ARRANGEMENTS

C. A. McKinney, *General Chairman*
J. van O. Weaver, *Vice-Chairman*

Registration: D. S. Cooper, *Chairman;* J. A. Bishop.
Publicity: J. A. Walsh, *Chairman;* H. J. Martyn.
Entertainment: A. B. Banowsky, *Chairman;* R. M. Spencer, C. R. Gardner, I. E. Rowe.
Finance: R. F. Taylor, *Chairman;* R. J. Salinger, R. K. Werner.
Tours: I. A. Naman, *Chairman;* W. L. Barnes, S. H. Green, R. J. Salinger.
Transportation: W. R. Etie, *Chairman;* F. A. Rodgers, Warren Earl.
Ladies: A. F. Barnes, *Chairman;* L. C. McClanahan, R. M. Spencer, J. D. Morrow.
Sports: G. D. Maves, *Chairman;* A. J. Mitchell, R. W. Kurtz, R. G. Lyford.
Banquet: A. M. Chase, Jr., *Chairman;* R. E. Tucker, B. J. Beaird, G. R. Rhine, M. L. Brown.

DEVELOPMENT OF INSTRUMENTS FOR THE STUDY OF AIR DISTRIBUTION IN ROOMS

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This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and conducted at the University of Illinois.

THE investigation of the distribution of air in rooms was undertaken to determine the air velocities and air temperatures existing at various points in a representative room, as affected by the location of the supply and exhaust openings, the temperature and velocity of the air at the supply openings, and the temperature outside of the exposed walls of the room. Incident to this investigation it has been necessary to develop some form of anemometer by means of which accurate observations of air velocities over a range of from 20 to 2000 fpm could be made without entering the room, and to perfect a wind tunnel for calibrating such anemometers. This paper constitutes a preliminary report on the construction of the test room, the construction of the wind tunnel and anemometers, and on results obtained from the calibrations of the wind tunnel and anemometers.

DESCRIPTION OF TEST PLANT

Test Room. A complete description of the room heating testing plant, containing the test room to be used for this investigation, has been given in a previous paper.[1] The test room is completely enclosed by a larger structure, having walls, floor and ceiling of 6-in. and 8-in. cork, which forms corridors on three sides of the room, and includes spaces corresponding to an attic and a basement. The walls enclosing the attic and basement spaces are insulated and

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[1] ASHVE RESEARCH REPORT No. 1163—Effect of Room Dimensions on the Performance of Direct Radiators and Convectors, by A. P. Kratz, M. K. Fahnestock, E. L. Broderick and S. Sachs. (ASHVE TRANSACTIONS, Vol. 46, 1940, p. 331.)
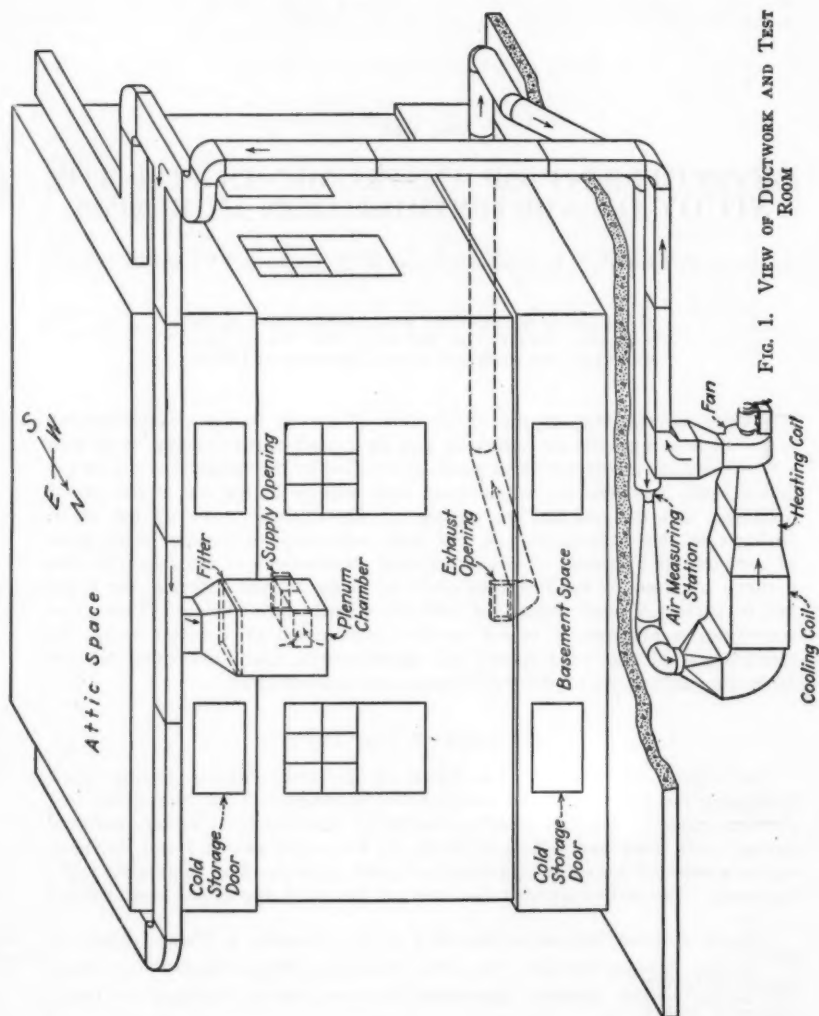
Fig. 1. View of Ductwork and Test Room

equipped with cold storage doors, and the spaces are provided with electric heaters, affording means of controlling the temperatures above the ceiling and below the floor of the test room. Refrigerating coils are placed in the front corridor and shielded from the front wall of the test room by means of a baffle, or radiation shield. The three corridors are also provided with electric heaters. The west corridor is equipped with cold storage doors so that it can be isolated from the other corridors. This arrangement, by using the proper combinations of refrigerating coils, electric heaters, and cold storage doors, makes it possible to operate the test room with either two or three walls exposed, and to maintain any desired temperature ranging from −5 F to +110 F in the outside corridors. The temperatures in the attic and basement spaces also can be controlled over a wide range.

The test room alone, without the enclosing corridors, is shown in perspective in Fig. 1. This room is 15 ft wide, 18 ft long, with an 8½-ft ceiling, and the walls are constructed of bevel siding, building paper, sheathing, 3⅝-in. studs, and plasterboard. The front wall contains two 3-ft x 4½-ft double-hung windows, and each of the two side walls contains a similar window and a 3-ft x 7-ft panelled wood door, with 27½-in. x 36-in. glass in the upper portion.

The test room, together with the corridors, the attic and basement spaces, are completely equipped with thermocouples and temperature recorders in order to permit all observations to be made without the necessity for entering the room and thus disturbing conditions at any time during a test. Thermocouples are located at various points in the inside and outside surfaces of the walls, floor and ceiling, as well as at various points in the air in the corridors, in the attic, and in the basement. Similar thermocouples used in connection with the anemometers will be located on a movable rack, spaced so that as the rack is progressively moved, a complete survey of the temperature of the air in the test room at points located approximately 2 ft from one another will be obtained. This rack will also carry the anemometers by means of which a simultaneous survey of the corresponding air velocities will be made.

As shown in Fig. 1, air from the air conditioning apparatus is supplied through a vertical riser to a horizontal belt duct located near the ceiling of the corridors, on three sides of the room. The air from this belt duct is admitted to a pendent plenum chamber, from which it is supplied to the supply opening into the room through a short length of horizontal duct extending into the plenum chamber. The plenum chamber can be moved to accommodate different locations of supply openings in the three walls, and the required connections can be made into openings provided in the bottom of the belt duct. The air from the exhaust openings is returned to the air conditioning apparatus through a 10-in. round duct, which can be adjusted in the basement space to accommodate the various required locations of the exhaust openings. Fig. 1 shows the arrangement with a supply opening 7 ft above the floor in the central axis of the front wall, and with an exhaust opening directly under the supply opening and in the center of the baseboard of the front wall. All portions of the ducts located in what may be cold spaces are insulated with 3-in. cork, while those portions not subjected to extreme temperatures are insulated with 1-in. rigid fiber insulation.

Air Conditioning Apparatus. The air necessary to maintain the required conditions in the room and the required velocities and temperatures at the

supply openings is conditioned by apparatus located as shown in Fig. 1. The room, duct work, and air conditioner, form a closed circuit. The air conditioning plant, as shown in Fig. 1, consists of appropriate heating and cooling coils, a fan, and a motor-driven condensing unit.

Cooling is accomplished by passing the air over extended surface coils through which chilled water is circulated. The temperature of the water entering the coil is regulated by means of a three-way mixing valve controlled by a thermostat located in the duct furnishing air to the supply openings. Chilled water is obtained from a reservoir containing a direct expansion coil con-
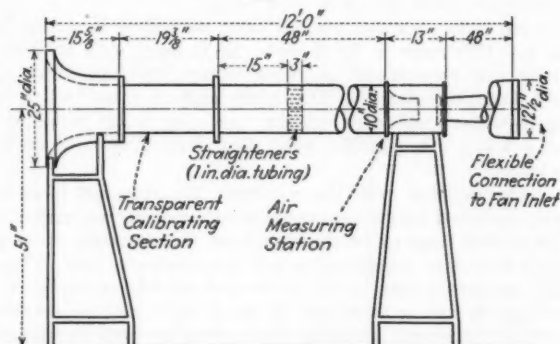
FIG. 2. WIND TUNNEL FOR CALIBRATION OF ANEMOMETERS

nected to the condensing unit. By using a modulating expansion valve, and by adjusting the speed of the compressor to give a capacity approximately equal to the cooling load on the test room, intermittent operation of the condensing unit may be avoided, and the water in the reservoir may be maintained at a constant temperature appropriate for use at the three-way mixing valve.

Heating is accomplished by passing the air over extended surface coils supplied directly with steam from the laboratory main. The air temperature is regulated by means of a valve in the steam line to the coil, which is controlled by a thermostat located in the duct furnishing air to the supply openings.

The amount of conditioned air circulated is measured by means of a calibrated nozzle located in the exhaust line just before the coils, as shown in Fig. 1.

TEST CONDITIONS

Cooling. The plant has been designed to provide for a maximum cooling load of 29,650 Btu per hour, based on an outdoor temperature of 110 F, an indoor temperature of 70 F, one air change per hour ventilation, six occupants using computing machines, and lights to the amount of 1350 w. Under these

conditions it will require 36.5 recirculations per hour, with the air leaving the supply openings at 50 F. By adjusting the size of the supply openings, velocities varying from 400 to 1200 fpm may be obtained.

Most of the studies will be made on a normal design load of 6,500 Btu per hour, based on an outdoor temperature of 95 F, an indoor temperature of 80 F, no ventilation except by infiltration, and no occupancy or lights. Under these conditions it will require 10 recirculations per hour, with air leaving the supply openings at 65 F. Velocities varying from 400 to 1200 fpm at the supply openings will be used.

Heating. The plant will provide for a heating load of 27,670 Btu per hour based on an outdoor temperature of 0 F and an indoor temperature of 75 F. Normally studies will be made with the air leaving the supply openings at a temperature of 135 F and velocities varying from 400 to 1200 fpm. Studies may be made with air leaving the supply openings at temperatures varying from 95 to 150 F.

INSTRUMENTS AND METHODS OF MEASUREMENT

Wind Tunnel and Measuring Instruments. The range of air velocities to be measured extends from velocities as low as 20 fpm to 2000 fpm. Such a range, as well as the fact that the anemometer measurements must be made without disturbing conditions in the test room, restricted the choice of an instrument to some type of thermal anemometer. The instrument finally selected was a heated thermocouple, somewhat similar to that developed by Hukill [2] for measuring thermal air currents of very low velocities. The principal advantage of such an anemometer over a hot-wire anemometer is that measurements may be made with a number of separate instruments and a single potentiometer, without the necessity of elaborate switching arrangements.

Fig. 2 is a line drawing of the apparatus for calibrating the anemometers. It consisted of a small draw-through wind tunnel, following the design used by Tuve.[3] The calibrating section was a 20-in. length of transparent duct, 10 in. in diameter, into which air flowed through a bell-mouthed entrance. The principal dimensions of this entrance, which had an elliptical approach, are given in Table 1. Following this section was a 48-in. length of 10-in. sheet metal duct leading to the measuring section, containing the measuring nozzle. From this latter section an expanding sheet metal section 48 in. long led to the inlet of the fan, which drew air through the tunnel. Since all air velocity determinations were to be based on flow measurements at nozzles calibrated by Pitot tube explorations, the measuring section of the wind tunnel was arranged so that the nozzles could be positively located with respect to the tunnel and that accurate pressure explorations could be made across the outlet of the nozzles. The expanding section, between the measuring section and the fan inlet, reduced the resistance of the system by allowing the conversion of dynamic pressure into static pressure to take place with a minimum loss due to shock and turbulence.

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[2] An Anemometer for Measuring Low Air Velocities, by W. V. Hukill. (*Refrigerating Engineering*, Vol. 28, 1934, p. 197.)
[3] ASHVE RESEARCH REPORT No. 1140—The Use of Air-Velocity Meters, by G. L. Tuve and D. K. Wright, Jr. (ASHVE TRANSACTIONS, Vol. 45, 1939, p. 645.)

The fan was belt driven by a compound wound direct current motor, so that partial control of the rate of flow in the tunnel was obtained by varying the speed of the motor. A sliding damper attached to the fan outlet provided additional control.

In Fig. 3 details of the measuring section are shown. This section was made of a piece of 10-in. steel pipe to which flanges were welded for connecting to the preceding and following sections of sheet metal duct. Two nozzles, one with a throat diameter of 4.250 in. and one with a diameter of 1.764 in. were required to make measurements throughout the chosen velocity range.

TABLE 1—DIMENSIONS OF NOZZLES WITH ELLIPTICAL APPROACH

	THROAT DIAMETER—IN. d	SEMI-MAJOR AXIS — IN. $a = d$	SEMI-MINOR AXIS — IN. $b = 2/3\,d$	LENGTH OF THROAT — IN. c
National Bureau of Standards D-1 Nozzle	1.764	1.764	1.176	1.764
National Bureau of Standards B-2 Nozzle	4.521	4.520	3.013	2.260
National Bureau of Standards C-2 Nozzle	5.003	5.000	3.333	2.500
University of Illinois Small Nozzle	1.764	1.764	1.176	1.764
University of Illinois Large Nozzle	4.250	4.250	2.833	2.125
University of Illinois Rounded Entrance	10.010	10.000	6.667	.... [a]

[a] Note: The entrance throat is continued into the calibrating section by a smooth connection.

Both nozzles were elliptical in shape, the larger being similar to the *B-2* and *C-2* nozzles used by Bean, Buckingham and Murphy in their investigation at the National Bureau of Standards,[4] while the smaller nozzle was identical with the *D-1* nozzle used in the same investigation. The principal dimensions of the nozzles are given in Table 1. The nozzles, which were cast of aluminum, were carefully machined and polished. They were located in the measuring section so that the end of either nozzle would lie in the same plane. Thus, the stream issuing from the nozzle could be traversed at the end of the nozzle by small exploring impact and static pressure tubes. Instead of measuring the drop in pressure, by means of pressure taps in the wall of the duct before and after the nozzle, the difference between the static pressure, as measured with a piezometer ring at the end of the nozzle, and the impact pressure, as measured with a stationary impact tube, was observed. The piezometer ring was made of ⅛-in. brass tubing which connected four 0.037-in. diameter holes drilled through the nozzle wall 0.25 in. from its end. The stationary impact tube also was made of ⅛-in. brass tubing, with dimensions similar to those of the ASHVE Standard Pitot tube. This tube was located at the axis of the nozzle and slightly more than 1 in. from the nozzle end.

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[4] Research Paper No. 49. *Bureau of Standards Journal of Research*, Vol. 2, 1929, p. 561.
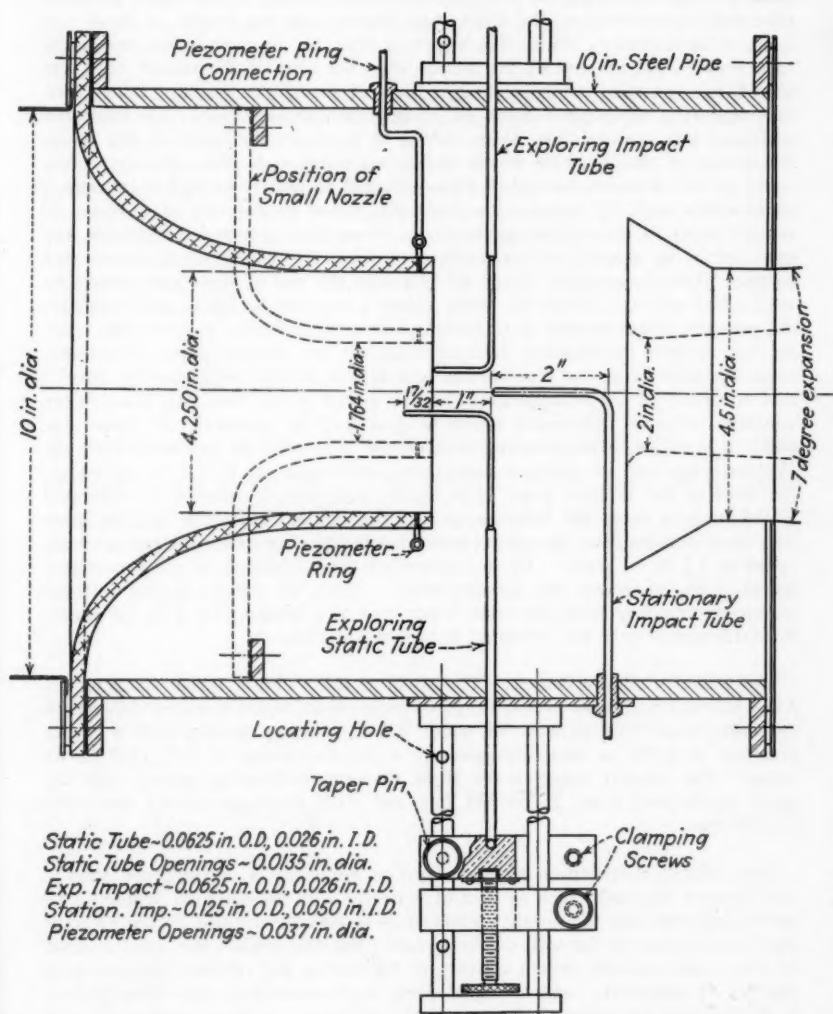
FIG. 3. AIR MEASURING STATION FOR WIND TUNNEL

The static pressure exploring tube was exactly similar to the Pitot tube and was so located that the four 0.0135-in. diameter holes, which formed the static pressure openings, were at the end of the nozzle. The impact pressure tube was also similar to the Pitot tube, except that the length of head was 1 in., or 16 diameters. With this length of head, the opening in the end of the tube would be at the end of the nozzle when the stem of the impact tube was placed in a position corresponding to that of the static pressure tube. The two exploring tubes were made of ⅛₆-in. brass tubing which, for increased stiffness, was sweated into ⅛-in. tubing 2 in. from the head of the tubes. By means of two movable heads sliding on guide rods, the exploring tubes could be moved across the end of the nozzle and accurately located at its center, or at either wall, by means of a taper pin placed in any one of a series of reamed holes in one of the guide bars. From these reference locations, the tubes could be moved to intermediate positions by means of a micrometer screw. Thus the position of the tubes across the end of the nozzle, could be established within ± 0.001 in. along either a vertical or horizontal diameter. All pressure measurements were made either with a Wahlen gage or with commercial inclined manometers, calibrated against the Wahlen gage. The apparatus for calibrating the manometers was similar to that suggested by Ower [5] and consisted of three large glass bottles placed in an insulated box. Very constant pressure differences could be produced by pumping air from one bottle to another. Direct comparison of the pressures, as measured with the Wahlen gage and an inclined manometer, was made up to 0.8 in. of water, the limit of the Wahlen gage. For higher pressures, a manometer calibrated to 0.8 in. was connected between two of the bottles, while the Wahlen gage was connected between the second and third bottle, thus extending the pressure range to 1.5 in. of water. By this procedure the calibration of the manometer up to 2 in. of water was accomplished. Since all errors in the inclined manometer reading were less than 1 per cent at a pressure of 2 in. of water, the calibrations were not extended to higher pressures.

Air Velocity Measurements. The small nozzle, with a throat diameter of 1.764 in., was calibrated over a range of piezometer-impact pressure differences extending from 0.04 to 9 in. of water while the large nozzle, with a throat diameter of 4.250 in. was calibrated for a pressure range of 0.25 to 7 in. of water. The velocity range in the 10-in. diameter calibrating section with the small nozzle was from 25 to 380 fpm and with the large nozzle from 360 to 2000 fpm.

The velocity distribution across a section following a convergence with a well-formed approach, such as that of a nozzle with an elliptical profile, is so nearly uniform that the nozzle coefficient is largely determined by the velocity distribution close to the wall of the nozzle. For this reason the usual method of Pitot tube traverse, which consists of measuring the velocity pressure in a number of concentric zones of equal area, is unsatisfactory unless the section is divided into an extremely large number of zones. A more convenient procedure is to space the pressure measurements inversely as the steepness of the velocity profile. The velocities, calculated from the velocity pressures,

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[5] Measurement of Air Flow, by E. Ower. *(Second Edition,* 1933, p. 187.)

are then weighted by the square of the corresponding radii, and the average velocity is their sum divided by the square of the radius of the section. This result may also be readily obtained by plotting the velocities against the square of their corresponding radii and measuring or calculating the area under the resulting curve.

With an exploring impact tube $\frac{1}{16}$ in. in diameter, and having an opening of 0.026 in., it was impossible to make satisfactory impact pressure measurements closer to the wall than 0.015 in. Even at greater distances from the wall, the readings were rather questionable, due to the extremely steep velocity profiles, especially with those which existed at the higher velocities. In order to extrapolate the velocity distribution in some systematic manner, from the region where reliable measurements could be made down to zero velocity at the wall, von Kármán's [6] analysis of turbulent flow at the wall of a pipe was applied to the flow at the nozzle wall. First the thickness of the laminar boundary was calculated from the relation

$$\delta = \frac{767.5\,R}{\left(\dfrac{RU_0}{\nu}\right)^{7/8}} \dotfill (1)$$

in which δ is the thickness of the laminar layer in inches, R is the radius of nozzle throat in feet and $\dfrac{RU_0}{\nu}$ is Reynolds' number based on the center velocity, U_0, in fps and the characteristic length R. The ratio of the velocity at the laminar boundary, U_δ, to the center velocity was found from

$$\frac{U_\delta}{U_0} = \frac{2.11}{\left(\dfrac{RU_0}{\nu}\right)^{1/8}} \dotfill (2)$$

and the velocity in the laminar boundary layer was assumed to vary linearly from U_δ to zero velocity at the wall. Within the turbulent boundary layer, the velocity was represented with a fair degree of accuracy by the dimensionless relations:

$$\sqrt{\frac{U}{\dfrac{\tau_0}{\rho}}} = 11.0 \log \sqrt{\frac{\tau_0}{\rho} \cdot \frac{y}{\nu}} \dotfill (3)$$

for the small nozzle and

$$\frac{U}{\sqrt{\dfrac{\tau_0}{\rho}}} = -2.2 + 13.0 \log \sqrt{\frac{\tau_0}{\rho} \cdot \frac{y}{\nu}} \dotfill (4)$$

for the large nozzle. In these expressions, τ_0 is the constant shearing stress in the laminar region found from

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[6] Fluid Mechanics for Hydraulic Engineers, by H. Rouse. (*First Edition*, 1938, p. 238.)

$$\sqrt{\frac{\tau_0}{\rho}} = 0.0862 \ U_\delta \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots (5)$$

ρ is the mass density in slugs per cubic foot and ν the kinematic viscosity in feet [2] per second, while y is the distance from the wall in feet. The constants in Equations (3) and (4) differ considerably from those in the von Kármán equation, the slope in the latter equation being approximately half that of either of the former. However, such results were to be expected, since the conditions of flow in the turbulent boundary layer in the nozzles correspond to that in the transition region between the laminar boundary and the turbulent boundary in fully established turbulent flow in a pipe.

The agreement between the measured velocities and those calculated on basis of the von Kármán analysis was best at the lower velocities and thicker boundary layers, when measurements could be made down to and, in a few tests, into the laminar layer. For this reason the method was used to obtain the velocity distribution in the boundary region for all of the nozzle explorations. From the complete velocity profile the average velocity and rate of discharge could be found as previously described.

Since the discharge coefficient of a nozzle is the ratio of the actual discharge to some theoretical or ideal discharge, if coefficients from different investigations are to be compared, they must be calculated on the basis of the same ideal discharge. When the drop in static pressure across the nozzle is measured, the theoretical discharge is usually that corresponding to frictionless, adiabatic flow for the given pressure differential. But if the differences in static pressures are determined by measurements at taps, located in the wall of the pipe, up stream and down stream from the nozzle, the resulting discharge coefficients are not strictly comparable unless the pressure measuring taps have identical locations and the conditions of flow, both before and after the nozzle, are duplicated. The discharge coefficients for the nozzles used in the present investigation were calculated on the basis of the discharge corresponding to a uniform velocity equal to the actual center velocity, as measured with the exploring impact and static pressure tubes. The center velocity, determined in this manner, should be very nearly equal to that of frictionless adiabatic flow for the static pressure differential which actually exists at the center of the nozzle. However, due to variations in static pressure which may exist across the stream issuing from the nozzle, the center velocity may differ slightly from the frictionless adiabatic flow velocity, calculated from the pressure differential measured by wall pressure taps.

Comparison of nozzle discharge coefficients is facilitated by plotting them against Reynolds' number as shown in Fig. 4. The plotted points are the coefficients for the two nozzles under discussion, while the broken lines represent the coefficients for two National Bureau of Standards [7] nozzles, one slightly larger than the large nozzle and the other identical with the small nozzle. From these curves, it is evident that the coefficients for the nozzles used in this investigation agree with the corresponding Bureau of Standards coefficients well within the limits of experimental error. The lower full line in Fig. 4 represents the average coefficients from a large number of tests by different investigators, using different fluids and various types of nozzles and

[7] Loc. Cit. Note 4.

orifices with gradual approach transitions, as given by Stewart and Doolittle.[8] They state that coefficients, obtained from various sources, showed such lack of agreement, particularly at flow with small Reynolds' numbers, that a curve which represents average values only, can be drawn.   Such lack of agreement is, at least partly, due to the effect of the type of approach used and to the location at which pressure measurements were made; because the agreement between the coefficients from the present investigation and those given by
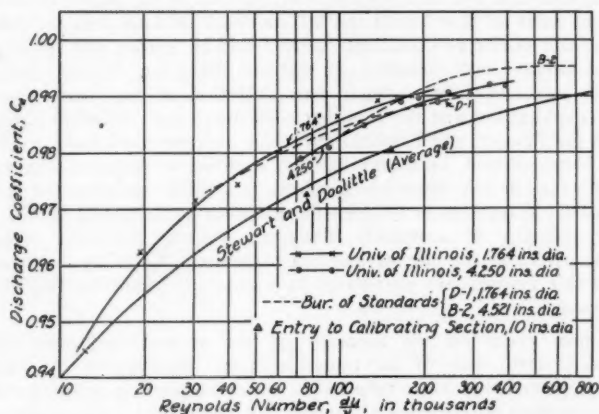
Fig. 4.   Comparison of Nozzle Discharge Coefficients on the Basis of Reynolds' Number for Air at 70 F and 29.92 in. hg

the National Bureau of Standards investigation indicates that the coefficients of similar nozzles are reproducible within ±0.5 per cent.

With nozzle coefficients based on center velocities, the average velocity, or the discharge from a nozzle, cannot be determined unless the relation between the center velocity pressure and the pressure drop across the nozzle has been established.   Even the pressure difference between a central impact tube and a piezometer ring at the wall of the nozzle does not give the true center velocity pressure on account of the variation in static pressure across the stream as it issues from the nozzle.   It was necessary, therefore, to find the ratio between the true center velocity pressure, as measured with the exploring impact and static pressure tubes, and the pressure difference between the stationary impact tube and the piezometer ring.   For the small nozzle, this ratio was found to vary from 0.992, at a piezometer pressure of 0.2 in. of water, to 0.994 at a pressure of 6 in.   Over the same piezometer pressure range, this ratio for the large nozzle was found to vary from 0.980 to 0.989.

The variation of static pressure across the stream at the end of the nozzle is probably the result of the variation in the centripetal force which must be exerted to produce flow along paths of varying curvature into the throat of the nozzle.   This change in static pressure has been observed by other investi-

[8] Fluid Flow Measurement, by F. C. Stewart and J. S. Doolittle.   (*Instruments*, Vol. 12, 1939, p. 175.)

gators [9] and has been discussed analytically by Kretzschmer [10] as a problem in potential flow. An attempt to predict the magnitude of such pressure variation, on the basis of the dynamics of flow, was unsuccessful, but the problem is still under consideration.

The average velocity in the 10-in. diameter calibrating section is related to the average nozzle velocity by means of the community equation. However, in calibrating small anemometers, the center velocity at the calibrating section must be known and this requires the determination of the discharge coefficients for various rates of flow out of the bell-mouthed entrance into the calibrating section. Such discharge coefficients were found by impact and static pressure tube explorations, with velocities of 890 and 1800 fpm through the section. The small exploring tubes, used in the calibration of the nozzles, were used for these explorations and the resulting coefficients were found to agree rather well with the Stewart and Doolittle curve for average coefficients, as shown in Fig. 4. Extrapolation, by means of such a curve, to obtain discharge coefficients with flow at low Reynolds' numbers, is a rather questionable procedure, in view of the disagreement between the available experimental results. However, the difficulty of accurately measuring velocity pressures under these conditions precludes the use of any other method. Certainly coefficients obtained in this manner are preferable to a constant coefficient for the entire velocity range.

As a final check on the accuracy of the several coefficients involved, comparisons were made of the center velocity, measured at the calibrating section by means of a Pitot tube, with the corresponding velocity calculated from the pressure difference at the nozzle. A summary of the results of these measurements is given in Table 2. As indicated in this table, velocity pressures at the calibrating section were measured with an ASHVE standard Pitot tube and with the $\frac{1}{16}$-in. exploring tubes, using both the Wahlen gage and a calibrated inclined manometer. For velocities greater than 750 fpm, the velocity, as measured with the Pitot tube, was consistently from 1 to 1.5 per cent higher than that found from the nozzle measurements. At lower velocities, the Pitot tube measurements became increasingly uncertain, because of the effect of pressure fluctuations when very small pressure differences were being measured. This condition was so pronounced when measurements were made using the small nozzle, that no consistent results could be obtained.

Although a number of possible reasons for this disagreement between the Pitot tube and nozzle measurements were considered, as for instance the effect of compressibility in the flow equation and in the continuity equation, and the effect of pressure fluctuations on the Pitot tubes and manometers; none seemed sufficient to account for an error of this magnitude. It was finally concluded that the most likely source of such an error was in the determination of the discharge coefficient for the calibrating section and, until explorations can be made at this section with some form of small thermo-anemometer, no further conclusions are possible.

Heated Thermocouple Anemometer. Some of the more desirable character-

[9] Nusselt. Zeit, F. Flugtechn., Vol. 6, 1915, p. 179. Jakob and Erk, V. D. I. Forschungsheft, 1924. No. 267.
[10] Stromungsform u. Durchflusszahl d. Messdrosseln, F. Kretzschmer, Forschungsheft, 1936, No. 381.

Table 2—Comparison of Velocities in Calibrating Section from Pitot Tube and Nozzle Measurements

Pitot Tube Size and Type of Manometer	Piezometer, Impact Pressure Difference In. of Water	Center Velocity at Calib. Section, Calculated Fpm	Pitot Tube Velocity Pressure at Calib. Section In. of Water	Center Velocity at Calib. Section (Pitot Tube) Fpm	Velocity Difference in Percent of Measured Velocity
$\frac{5}{16}$" ASHVE Standard Wahlen Gage......	0.30	412.7	0.0104	416.4	0.89
$\frac{5}{16}$" ASHVE Standard Wahlen Gage......	0.80	671.1	0.0274	686.1	2.19
$\frac{5}{16}$" ASHVE Standard Wahlen Gage......	1.20	822.2	0.0404	836.4	1.70
$\frac{5}{16}$" ASHVE Standard Wahlen Gage......	1.80	1005	0.0601	1014	0.89
$\frac{5}{16}$" ASHVE Standard Wahlen Gage......	2.20	1113	0.0735	1124	0.98
$\frac{5}{16}$" ASHVE Standard Wahlen Gage......	2.95	1287	0.0983	1299	0.92
$\frac{5}{16}$" ASHVE Standard Inclined Gage......	1.80	1012	0.0625	1023	1.08
$\frac{5}{16}$" ASHVE Standard Electronic Gage....	1.80	1012	0.0632	1029	1.65
$\frac{1}{16}$" Exploring Tube Wahlen Gage......	1.80	1012	0.0626	1027	1.46

istics in an anemometer for the proposed study of air distribution are the following:

1. The instrument should be remote reading, so that air velocity measurements can be made without the observer entering the room.
2. The instrument should measure the velocity across a section of approximately the same proportions as that measured by the standard Pitot tube.
3. The range of velocities, throughout which the instrument is sensitive, should be from 20 to 2000 fpm.
4. The instrument should have a minimum directional response.
5. Duplication of the instrument should be relatively easy.

In Fig. 5 is shown the final design of heated thermocouple, which was developed to meet the requirements specified. The wiring diagram of the thermocouple and heater circuits is also shown in Fig. 5.

The frame of the anemometer consists of three $\frac{1}{2}$-in. pieces of No. 24 brass or steel spring wire, soldered together so as to be mutually perpendicular. One of the wires was cemented into a hole drilled in the end of a piece of round rawhide belt lacing, which forms the support for the anemometer. The thermocouples were made from No. 34 B. & S. gage copper and constantan

wire. Enamel insulated wire was used and, after the lap soldered couples had been made, the wire was given two coats of insulating lacquer. The electric heater was made of No. 34 B. & S. gage enameled, constantan wire, which

WIRING DIAGRAM
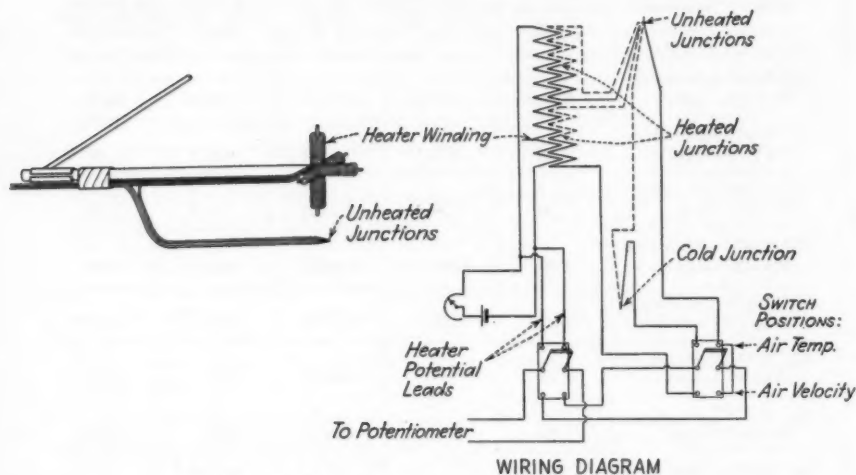
Fig. 5. Heated Thermocouple Anemometer

was also given two coats of insulating lacquer before being wound. The heater and thermocouple wire were wound side by side on the wire prongs, 28 turns on the center prong and 18 turns on each of the side prongs. Two heated and two unheated thermocouples were used. These couples were connected in series, in order to increase the electro-motive force due to the temperature difference between the heated and unheated thermocouples. As indicated in the wiring diagram, Fig. 5, one of the unheated junctions of the anemometer was soldered directly to a copper-constantan junction made of No. 24 wire, while the other unheated anemometer junction was cemented to, but insulated from, the large junction by means of insulating lacquer. The anemometer electro-motive force was measured between the copper lead from the anemometer junctions and the copper lead from the No. 24 wire junction. The electro-motive force between the No. 24 wire leads indicated the temperature of the air close to the point where the velocity measurement was being made.

The anemometer heater was supplied with current from several dry cells and a rheostat was used to control the current. The potential drop across the heater was measured with the potentiometer which was used in measuring the electro-motive force of the anemometer thermocouples.

Calibration of Heated Thermocouple Anemometer. The calibration curves in Fig. 6 show the air velocity plotted on logarithmic coordinates against electro-motive force difference between heated and unheated anemometer junctions. From these results it is clear that, although velocity measurements could

be made over the entire range, with the same heater voltage, it was desirable to use three or four different voltages throughout the range. With potentials of 0.75, 1.00 and 1.5 volts it was possible to measure velocities from 20 to 2000 fpm with satisfactory precision and without excessive temperature differences between the heated and unheated thermocouples. If the velocity range was not too great for a given voltage, the calibration curves were found to be straight lines having a slope of approximately 2.6. The wire prongs, which extended beyond the heater coils, were trimmed in order to balance the heat loss; so that the electro-motive force measured when the axis of the anemometer was parallel to the direction of flow agreed with the electro-motive force measurement when the flow was perpendicular to the axis. Plotted points in Fig. 6 indicate that the difference in electro-motive force in these two positions may be reduced to the order of magnitude of the random experimental errors.
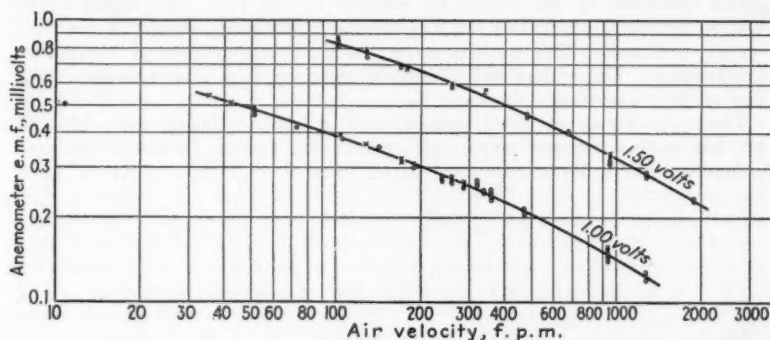
FIG. 6. CALIBRATION CURVES FOR HEATED THERMOCOUPLE ANEMOMETER

However, it should be noted that, due to the steepness of the calibration curves, the velocity measurements with the heated thermocouple anemometer have only about half the accuracy of the corresponding electro-motive force measurements.

SUMMARY OF RESULTS

This paper describes a wind tunnel which was constructed for the calibration of anemometers over a velocity range from 20 to 2000 fpm. All velocity measurements are based on flow measurements with two elliptical approach nozzles, which were calibrated by means of small exploring impact and static pressure tubes. Although nozzle discharge coefficients from different investigations cannot be compared, unless they are calculated on the basis of the same ideal or theoretical discharge, the coefficients from this investigation, based on actual center velocities, agree within ±0.5 per cent with the coefficients for similar nozzles found at the National Bureau of Standards and based on the discharge corresponding to adiabatic flow.

The discharge coefficients for the calibrating section of the wind tunnel were found from Pitot tube measurements in the velocity range where such measurements could be made with sufficient accuracy. The coefficients for velocities below this range were found by extrapolation on the basis of Rey-

nolds' number, using a curve representing nozzle discharge coefficients, with which the measured coefficients were in good agreement at the higher velocities.

The construction of a heated thermocouple anemometer is described which has certain definite advantages where low velocities are to be measured in a closed space without disturbance due to an observer. The anemometer may be made practically free from directional effects and, by the use of suitable heating currents, it is possible to measure air velocities with satisfactory precision over a wide range, without excessive temperature differences between the air and the heated thermocouple.

COOLER FOOTCANDLES FOR
AIR CONDITIONING

By W. G. DARLEY,* CLEVELAND, OHIO

IN a fundamental paper [1] presented in 1938, some comment was made on the fluorescent lamp, a new light source then under development which was not commercially available; however, data were presented covering an experimental 15-w (watt) white fluorescent lamp. In the intervening $2\frac{1}{2}$ years the 15-w white Type F lamp has not only been made available, but has been improved to a point where its efficiency is 30 per cent higher than the experimental lamp which was described. Furthermore, higher wattage lamps (20, 30, and 40 w) are now listed which have even higher light outputs per watt input. For instance, the 40-w white fluorescent lamp with an efficiency of 53 1/w (lumens per watt)† is over 75 per cent more efficient as a light producer than the experimental lamp.

These developments have greatly broadened the field of application for this type of source beyond that suggested in the previous paper. No longer are the fluorescent lamps considered only as better tools than the lumiline lamps for use in auditoriums, theaters, restaurants, etc. They have taken their place with Type C (gas-filled tungsten-filament) and Type H (mercury) lamps in the lighting engineers' kit as sources of many uses. Already approximately four million are in use.

During the past $2\frac{1}{2}$ years there has also been a considerable amount of new research on seeing as it is influenced by lighting.[2, 3, 4, 5, 6] In addition to the technical reports of these data the continuation of the *better light—better sight* movement and the activity on light-conditioning are bringing the researches to the attention of the public without undue delay. More information is now available on the effect of better lighting on selling in the store [7] and production in the factory.[8] The cumulative effect of all these factors has been to improve lighting practice both as regards quantity of light (footcandles) and quality of lighting (considering glare, distribution, diffusion, shadows, etc.). For instance, in 1938 the recommended minimum standards of illumination for private and general office classifications of *no close work* and *close work* were 10 and 20 ft-c. For some time the accepted minimum standards for such work

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* Nela Park Engineering Dept., General Electric Co.
[1-8] Numbers refer to Bibliography.
† Lamp alone, at 100 hours.

have been 20 and 30 ft-c respectively. Research previously mentioned indicates that for many common office tasks 100 ft-c or more are really necessary for true seeing ease.[9]

Of course, it has been possible to achieve the higher footcandle values in isolated instances by means of general plus supplementary lighting. Fig. 1 illustrates this method. In this case, 30 ft-c of general illumination from indirect luminaires are supplemented with about 80 ft-c from the directional unit located over the desk. This method, however, is only suitable where the areas or materials to be lighted are permanently fixed in place; and even then the quality of the lighting leaves something to be desired as the brightness of the supplementary source can cause annoying reflections. It is found, there-

Fig. 1. Private Office with Location of Desk Fixed, the Directional Supplementary Lighting Unit Mounted at Ceiling Adds 80 ft-c to the 30 ft-c from General Lighting Units to Give Over 100 ft-c on the Work

fore, that the solutions for most office lighting problems (solved with Type C lamps) have been large-area, low-brightness general lighting systems. One of the most common systems of this type is that using conventional indirect luminaires in conjunction with a matte white ceiling (Fig. 2).

One of the hindrances to the utilization of higher footcandles of general lighting with Type C lamps has been the complex psycho-physiological reaction which causes us to brand an installation as *too hot*. Three of the components which influence this reaction are: (1) total heat liberated in a space; (2) radiant heat which, for the purposes of this discussion, may be stated in terms of the amount of radiated energy per unit of light; and (3) psychological reaction to the color and brightness of the source, to the color and brightness of the surroundings, etc.

Good commercial general lighting practice with Type C lamps today (20 to 30 ft-c) involves the use of a wattage of the order of 4 to 6-w per square

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[9] See Bibliography.

foot. From the data presented in 1938 it is indicated that during a 7-hour period this loading would raise the temperature in a completely unventilated office (17 ft by 17½ ft, two windows) only 3 to 4½ F. Since with the two windows and transom open on a typical summer day the rise would be much less, about 1 F, it appears that artificial ventilation could readily provide comfortable conditions with the above loading from the standpoint of the total heat effect.

At the same time it appears that 20 to 30 ft-c from Type C sources should cause little complaint from the standpoint of radiated heat. For instance, an experiment was conducted recently [10] in which subjects were blindfolded and had light directed upon their foreheads. The light was interrupted four times per minute. The average minimum illumination from bare 100-w tungsten-filament lamps which the subject could detect by a barely noticeable

FIG. 2.   TYPICAL INDIRECT LUMINAIRES UTILIZING SILVERED-BOWL
LAMPS WITH FLAT WHITE CEILING

increase in the sensation of warmth was 125 ft-c (range 75 to 175 ft-c). It appears quite reasonable to suppose that had the light been directed upon the forehead without interruption for a longer period (30 min or more), the illumination detected might have been lower, because when radiant energy is absorbed by the skin it has the same heating effect on the skin as on any other object except that the skin has one added method of disposing of some of the heat, that of the bloodstream being able to carry it away. Doctors Winslow, Herrington, and Gagge [11] in a series of elaborate experiments determined that the most pleasant condition was registered when the skin temperature is around 90 F. At 5 F above or below this the subject complains of warmth or cold. It is logical to assume, of course, that the amount of variation required to cause complaint would differ for individuals and that some degree of discomfort must be present as soon as the temperature varies from the most pleasant condition even though the point is not reached where the subject is conscious of the discomfort and complains. However, from these tests it does seem obvious that 20 or 30 ft-c should not cause any annoyance because of radiant heat.

[10-11] See Bibliography.

The foregoing considerations leave psychology to bear quite a bit of the burden of guilt for producing complaints of heat. The *blindfold test* previously mentioned is fundamental for it suppresses psychological factors. One has often experienced the sensation of heat in interiors lighted with artificial illuminants and therefore may expect artificial lighting to be warm, having burned fingers many times changing lamps. Heat is associated with the *warm* color of light from Type *C* lamps, and naturally the more light the hotter it seems.

Lighting practice has taken these factors into account in recent higher footcandle installations using Type *C* lamps. For instance, reductions in total heat per footcandle have been obtained by developing lighting methods which provide higher utilizations, *e.g.*, more footcandles per watt expended. One of the most popular methods of this type is that known as coffer lighting, illus-

FIG. 3. COFFERS 4 FT X 4 FT RECESSED INTO CEILING 15 IN. TO PROVIDE HIGHER UTILIZATION THAN OBTAINED FROM CONVENTIONAL INDIRECT SYSTEMS

trated in Fig. 3.[12] This method involves the use of large coffers (some 4 ft x 4 ft, and 15 in. deep) recessed in the ceiling and equipped with louvers to shield the eyes from the brightest portion of the silvered-bowl lamps used. In effect, the coffer is a very large direct lighting unit. Besides having a higher utilization, the coffer method results in the highest ceiling brightness being hidden from normal view, thus reducing the psychological sensation of heat.

The matter of radiant heat has also come in for its share of study. As a result of cooperation between paint manufacturers and illuminating engineers progress has been made on the development of a new type of paint for use on light-reflecting surfaces. Paints have been made which reflect 75 per cent of the light (visible radiation) emitted from Type *C* lamps but which reflect only 30 per cent of the total energy radiated from these lamps.[13]

One of the most interesting approaches toward overcoming the undesirable psychological sensation of heat from too high a brightness in the field of view has been the development of a metal ceiling with an aluminum finished surface having ridges of a special contour which reflect the light rays in such a manner

12-13 See Bibliography.

that a larger percentage is directed downward, reflections near the horizontal being minimized [14] (Fig. 4). As a result, the brightness of this ceiling, when viewed from a distance, appears much less than that of the usual flat white ceiling receiving the same light flux. Thus for the same number of foot-candles on the work plane the sensation of brightness is materially reduced, the color of the ceiling changes from the warm white obtained with a flat white ceiling to a neutral gray, the result being that the psychological reaction is decidedly on the cool side.

It appears, therefore, that the way is still not closed to the attainment of higher values of *cool footcandles* with Type C lamps.
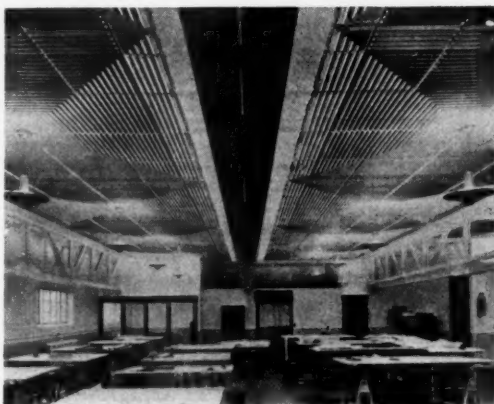
Fig. 4. Formed Metal Ceiling Painted with Alumi-num Paint Directing Light Downward Rather than Diffusing It in all Directions, as in Flat White Ceiling

Now take into consideration fluorescent lighting and its effect on air condi-tioning as regards (1) total heat, (2) radiant heat, and (3) psychological reactions.[15]

Total Heat

The Type F lamp differs radically from the Type C and Type H lamps in that instead of being concentrated and having a high brightness (maximum for 500-w inside-frosted Type C lamp—290 candles per square inch) it is extended and has a relatively low brightness (maximum for 40-w white Type F lamp—4 candles per square inch). While the source brightness of the fluorescent lamps, when compared to that of the filament lamps, is thus hardly more than a glimmer, it is still high enough to be a limitation to the indiscriminate use of fluorescent lamps for direct and direct-indirect lighting in commercial areas. It is felt, for instance, that the brightness of the 1-in. (T-8) Type F lamps

14-15 See Bibliography.

is high enough to exclude their use in any system which permits direct viewing of the lamps by or direct specular reflection of the lamps to the eyes of the observer. On the other hand, for the majority of commercial uses the brightness of the 1½-in. (T-12) Type F lamp is judged to be tolerable from the standpoint of reflected brightness, particularly if footcandles are high (50 ft-c and up). At the same time, the brightness of the T-12 lamp is higher than one would care to tolerate in the normal line of vision for extended periods.

There are two viewing directions for which the fluorescent lamps can be shielded: one for viewing crosswise, or normal, to the axis of the lamp (Sketch
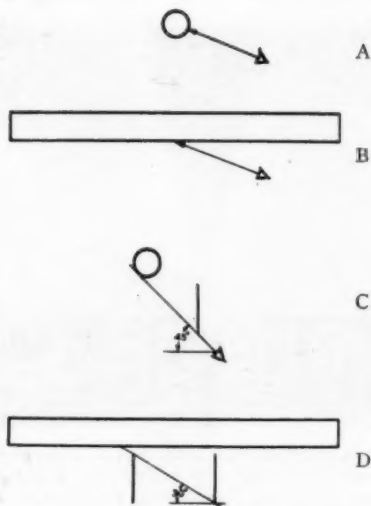
FIG. 5. MAXIMUM FLUORESCENT LAMP BRIGHTNESS IS FOUND AT THE NORMAL OR CROSSWISE VIEWING ANGLE, A

A, Fig. 5); and the other for viewing along the axis of the lamp (Sketch B). The maximum brightness is encountered when viewing crosswise. Observation indicates that for viewing in this direction if the T-12 lamps is shielded to a 45 deg angle below the horizontal (Sketch C), a good degree of comfort will prevail; that for viewing along the axis, a shielding angle of at least 30 deg is desirable (Sketch D). The shielding media can be either opaque or translucent)—if the latter, their brightnesses should be of a low order, depending somewhat upon the system. Of course, more shielding than indicated should provide greater freedom from the possibility of annoying direct glare. In other words, if care is taken, the T-12 Type F lamp may be used to provide direct and direct-indirect lighting with an acceptable degree of comfort. Figs. 6, 7, and 8 illustrate such typical fluorescent lighting methods. These systems result

Fig. 6. Six 40-w White Fluorescent Lamps Per Outlet in Open Top Direct-indirect Luminaires. "Eggcrate" Louvers Provide Shielding of the Lamps from Below. Forty-five Footcandles Average in Service are Obtained in the Room

in higher utilization of the light generated than the more commonly-used, comfortable forms of filament lighting.

Thus there are often two factors which tend to reduce the total heat per

Fig. 7. Direct Lighting Units Mounted at Ceiling, Each Accommodating Six 40-w White Fluorescent Lamps, Provide 40 ft-c Average in Service in Office

footcandle with Type *F* lamps: (1) higher lamp efficiency, and (2) higher utilizations of light. For instance, Table 1 shows the relative watts (and Btu) per square foot required to obtain 50 ft-c average in service in typical large and small offices. Since the auxiliaries for the Type *F* lamps are usually located in the same room with the lamps, the wattage (heat) losses in the auxiliaries are included. For the indirect and coffer filament systems and the direct-indirect and troffer fluorescent systems, it is apparent that for the same dissipation of total Btu per square foot, it is possible to obtain over twice as many footcandles with fluorescent lighting as with filament lighting. Considering an extreme case, an allowance of 5-w per square foot in an office 14½ ft x 17½ ft (11½-ft ceiling) would work out as follows:

1. Conventional indirect (filament lamps)...................................16 ft-c
2. Coffer (filament lamps)..27 ft-c
3. Troffer (fluorescent lamps)..64 ft-c

Fig. 8. Troffer System of Fluorescent Lighting, for Use with Sound-conditioning. Continuous Rows of Troffers, on 3-ft Centers, Accommodating 40-w White Fluorescent Lamps Provide an Illumination of 45 ft-c

Thus, if 5 w per square foot is a practical load for which to air condition today, desirably high footcandles are immediately available with Type *F* lamps.

Radiant Heat

While the reduced total heat per footcandle with fluorescent lighting is significant, it is in the lower radiant energy per lumen that the greatest gain lies.

One is not conscious of any great amount of heat or heating effect from the hundreds of footcandles of light coming from the sky through a window or in the shade outdoors. Daylight footcandles are relatively cool, primarily because the high temperature of the sun results in much less invisible energy accompanying each lumen than is the case for most artificial illuminants. This is analogous to a high luminous efficiency for a lamp. In addition to this primary cause of cool daylight footcandles, the water vapor in the atmosphere absorbs a considerable percentage of the invisible infra-red energy accompany-

ing the light. The result is high illumination without perceptible or undue discomfort due to heating effect. Type *F* lamps provide *daylight* footcandles, partly due to the increase in luminous efficiency and partly due to the methods of light-production involved. A study of Figs. 9 and 10, illustrating the energy distributions of a 500-w Type *C* lamp and a 40-w white Type *F* lamp respectively, shows that on the basis of total radiated energy the radiated watts per lumen for the filament lamp are 0.045, for the fluorescent lamp are only 0.0087. Thus 250 ft-c from fluorescent lamps are no warmer than 50 ft-c from filament lamps!

Besides confirming that the heating effect per footcandle for light from 40-w white Type *F* lamps is only about one-fifth that of light from large Type *C* lamps, Table 2 reveals other surprising results. For instance, the heating effect

TABLE 1—Approximate Watts Per Square Foot for 50 Footcandles, Average
in Service for Various Lighting Systems

SYSTEM	ROOM SIZE[a]				UTILI-ZATION FACTOR	WATTS PER SQ FT[b]	BTU PER SQ FT PER HOUR LAMP OPERATION[b]
	Width		Length				
	Ft	In.	Ft	In.			
FILAMENT LIGHTING							
Conventional Indirect......	14	6	17	6	0.25	15.4	52.5
Conventional Indirect.......	50		70		0.44	8.7	29.8
Coffer....................	14	6	17	6	0.36	9.3	31.6
Coffer....................	50		70		0.52	6.4	21.9
FLUORESCENT LIGHTING							
Direct-Indirect............	14	6	17	6	0.31	5.3	18.1
Direct-Indirect............	50		70		0.46	3.6	12.2
Troffer...................	14	6	17	6	0.39	3.9	13.4
Troffer...................	50		70		0.55	2.8	9.5

[a] All ceiling heights 11 ft 6 in. Average reflection factors: Ceiling 75 per cent; sidewalls 30 per cent average.
[b] Includes losses in ballasts of Type *F* lamps—40-w white Type *F* lamps over-all efficiency 43½ lumens per watt. Type *C* lamps at 20 lumens per watt.

of light from the 40-w white Type *F* lamp is only about twice that of the *coolest* daylight *e.g.*, skylight through a closed window. When a thin piece of ordinary glass is interposed between the lamp and the place where the foot-candles are measured, the heating effect per footcandle is even less than that of the *coolest* daylight!

To find out whether or not the energy distribution data (Figs. 9 and 10) and the energy-footcandle measurements (Table 2) really applied to human beings, the blindfold test was repeated using a bare 15-w daylight Type *F* lamp (30 lumens per watt). With the light directed upon the forehead interrupted four times per minute the average minimum illumination which the subjects detected by a barely noticeable increase in the sensation of warmth was 600 ft-c (spread 500 to 700 ft-c). This compares to the average of 125 ft-c previously given for the 100-w Type *C* lamp. Extrapolating, it is found that for the more efficient 500-w Type *C* and 40-w white Type *F* lamps the ratio would be about 160 to 900 ft-c. Thus these more practical tests show that
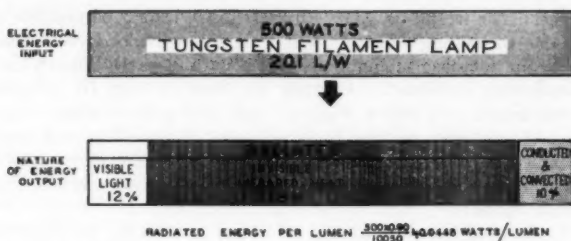
Fig. 9. Distribution of the Energy Emitted by a 500-w
Tungsten-filament Lamp [16]

the energy content of the light from the more efficient Type F lamps is only of the order of 1/5 to 1/6 that of the light from commonly-used Type C lamps. The blindfold test thus verifies with human skin and its sensory nerves the results obtained with the thermopile which measured the heating effect per footcandle. Approaching the coolness of natural daylight with artificial daylight is an achievement of very great significance and potentiality.

Psychological Reactions

Psychological reactions are determined in large measure by past experiences one way or another. Thus, to date the sensation of heat from artificial illumination in interiors has often been experienced and, although it is likely to be warm, the heat in footcandles from skylight through windows is not expected to be noticeable. It should also be noted that with daylight we are accustomed to hundreds of footcandles without an undue sensation of heat while artificial lighting has been felt to be warm at relatively low illuminations. Since both the white and daylight Type F lamps have color temperatures nearer to that of daylight than commonly-used Type C lamps, the light from the Type
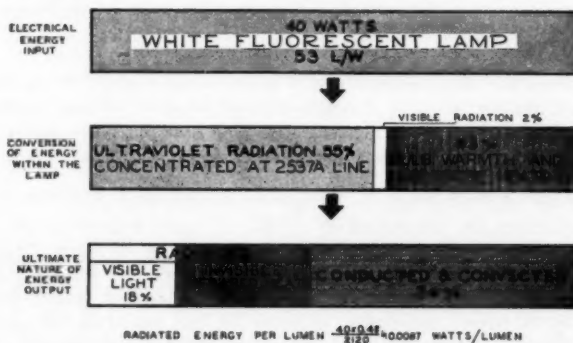
Fig. 10. Distribution of the Energy Generated and Emitted
by a 40-w White Fluorescent Lamp

[16] See Bibliography.

F lamps is definitely psychologically cooler than that from Type C lamps. Naturally the daylight Type F lamp rates considerably *cooler* on this basis than the white Type F lamp.

Within limits, a reduction in the area or degree of a brightness in the field of view results in a reduction in the psychological sensation of heat. In many of the systems utilizing the T-12 Type F lamps, the lamps are well shielded from the view of the room occupants but not from view by the work. (Figs. 6, 7, 8.) Thus the brightness of the work is increased without increasing the brightness of the luminaires or ceiling in the normal field of view. Naturally, the sensation of heat also depends somewhat upon the color and intensity of all surrounding brightnesses.

Summary

At the wattages per square foot for which it seems practical to air-condition today, the Type F lamps permit illuminations far in advance of those obtain-

TABLE 2—Approximate Relative Radiant Energy or Heating Effect Per Footcandle

Source	Relative Energy per Footcandle
40-w white Type F lamp (53 1/w)	1.00
40-w daylight Type F lamp (45 1/w)	1.25
40-w inside-frosted filament lamp (11.7 1/w)	6.6
100-w inside-frosted filament lamp (16.2 1/w)	5.7
200-w inside-frosted filament lamp (18.4 1/w)	5.1
500-w inside-frosted filament lamp (20.0 1/w)	4.5
Sunlight through open window	1.25
Sunlight through ⅛ in. glass window	1.04
Skylight through ⅛ in. glass window	0.67
⅛ in. of ordinary glass interposed between 40-w white Type F lamp and surface on which measurements were made [a]	0.44

[a] For filament lamps this would result in about a 20 per cent reduction in heating effect per footcandle.

able with Type C lamps because of their higher efficiency and because they can be used in systems having higher utilizations, such systems being made feasible by the lower brightnesses of the lamps. At these higher illuminations with Type F lamps there may be less heat effect per footcandle than for skylight—certainly less if the majority of the light passes through glass or plastic before reaching the room occupants. Furthermore, since the daylight and white lamps give psychologically cooler colors of light, we are indeed ready to enter the new era of a completely controlled environment—an environment with light, air and sound conditioning, an era of competing not with darkness but with daylight.

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DISCUSSION

EDWIN D. TILLSON** (WRITTEN): Some may be startled at what seems to them very high illumination values. Utility lighting men today are put to some confusion by prospects for relighting who confront them with their own (the salesmen's) recommendations of a year or so previously; for example, 25 ft-c incandescent (5 w per square foot) for an office installation (all on the authority of the design books, it should be noted). Now the prospect is asked to install 50 to 75 ft-c of fluorescent lighting in the same office.

However, the customer complains—"Your design book said 25 ft-c was *enough*. What excuse is there for three times this value now? If it was not necessary then it is not today."

It all gets back to what is *enough*. This same prospect can go to a dozen different places and buy typewriters for $50 per machine—then why pay $225 for an electric? And yet the electric machines are now seen almost everywhere. Similarly with automobiles, a $700 car will take us to as many places as fast and probably cheaper than a $1400 machine. Then is not the $700 car *enough?* Well, apparently many think otherwise.

The only place we might differ with the author is in some of the ways of getting the higher footcandles. For example, such as illustrated in Fig. 8 of the paper.

If we are out to reproduce natural daylight indoors, should we not endeavor to duplicate daylight brightness, diffusion, contrast and color as well as intensity? The

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writer in times past has used the following illustration. The blue vault of the heavens is our natural daylight source, whereas most artificial lighting in the past has been a dark void punctured with point sources of light on a regular spacing, vastly different from natural lighting you will concede, and so is the type of lighting shown in Fig. 8, where brightness contrasts of 100 to 1 have been measured. There are ways of installing fluorescent lamps to simulate natural lighting in a much better way than is being done with fluorescent lamps at this time. We trust that illuminating engineers will strive for the ideal in fluorescent lighting as they have always done with incandescent lamps in the past.

D. H. TUCK†† (WRITTEN): The subject of this paper is not apt because it is a statement of fact that the paper in itself does not prove. There is a great difference between lamps and footcandles on the work and the subject of the paper deals almost entirely with lamp operation and not so much with a system of illumination. The *blindfold test* referred to showed that for the average person the radiant heat associated with 125 ft-c of incandescent lighting would produce just a perceptible sensation of warmth and it is natural to assume that the footcandles could be increased considerably before the lighting could be classed as uncomfortable.

In a paper recently presented‡‡ before the *Illuminating Engineering Society* it was stated that from sales reasons the illumination in department stores should not be greater than approximately 20 ft-c. There is nothing in the author's paper to show that 20 ft-c are a factor in department store lighting from an air conditioning standpoint. For office lighting there is good reason to go as high as 50 ft-c because the 50 ft-c are measured on a horizontal plane which would mean approximately 25 ft-c on a vertical plane and assuming a uniformity ratio of 1.5 would mean a vertical illumination of 17 ft-c, actually on the work, which is sufficient for efficient office work. (The high footcandle values often referred to are misleading because they are fictitious footcandles and do not represent actual minimum footcandles on the work.)

The figures given in Table 1 are obviously shaded to favor fluorescent lighting. No mention is made in the table of direct-indirect filament lighting which would be comparable with direct-indirect fluorescent lighting. Also the values given for watts per square foot for the indirect filament lighting are unduly high, *e.g.* the 8.7 w per square foot could just as well be 5.3 w per square foot if efficient equipment had been figured, and would be 4.0 for a direct-indirect system comparable in quality to a direct-indirect fluorescent system. On the other hand, the watts per square foot figure of 3.6 for fluorescent direct-indirect is evidently low because as already pointed out in the paper, the luminaire should have the lamps shielded and when this is done the figure of 3.6 for the fluorescent system becomes 4.0.

Thus, it would seem that on the basis of equal footcandles, equal type of lighting (direct-indirect), and equal brightness that the total watts and therefore total heat is the same for both filament lamps and fluorescent lamps and that, inasmuch as radiant heat effect is of no importance under 125 ft-c, that there is no difference here that would affect air conditioning.

It is generally conceded that two footcandles of fluorescent are required for 1 ft-c of filament lighting for equal seeing. If subsequent scientific investigations bear out this belief, then from the standpoint of total watts or total heat the fluorescent lighting would suffer and we would have *hotter footcandles for air conditioning.*

AUTHOR'S CLOSURE: The practical analysis of footcandles recommendations contributed by Mr. Tillson is a very worthwhile addition to the paper.

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†† Electrical Engineer, Engrg. Dept., Holophane Co., Inc., New York, N. Y.
‡‡ October, 1940.

With reference to Mr. Tillson's comment concerning the type of lighting shown in Fig. 8, it is interesting to note that since this paper was presented aluminum and aluminum-finished troffers have been developed which provide greatly improved bright-contrast conditions in the normal field of view. So far no one lighting system has been perfect, and as illuminating engineers strove to reach the ideal with filament lighting, undoubtedly so will they strive for the ideal with fluorescent lighting.

Had Mr. Tuck perused the paper more carefully he undoubtedly would have come to the realization that the title is a statement of a fact which is proven conclusively by the paper. While I was interested in Mr. Tuck's statement concerning footcandles in department stores, I could not find the *Illuminating Engineering Society Transactions* referred to (October 1940). I do know, however, that there are numerous stores being lighted to well over 50 ft-c, some going even as high as 100 ft-c average maintained in service.

While I do not entirely follow Mr. Tuck's mathematics on the office lighting examples, or see why we should be primarily interested in vertical illumination in an office, I am in agreement that the footcandles measured on the horizontal do not represent actual footcandles on the work inasmuch as the body of the worker may cut out a third or more of the illumination at the work point.

The utilization factors used for the conventional indirect lighting in Table 1 are for units having outputs of 80 to 85 per cent, which is about tops as far as efficiency is concerned for such comfortable, low-brightness equipment, or for any other type of equipment for that matter. It is quite true that direct-indirect units for filament lamps, such as enclosing globes with parchment shades around them, provide lighting which is quite comparable to fluorescent direct-indirect lighting insofar as brightness and distribution are concerned. Since fluorescent lighting equipment of this type runs as high as 80 per cent output, the output of the filament units might be expected to equal but certainly could not be 100 per cent more efficient (to obtain a total output of 175 per cent) than the fluorescent luminaire which is what would be required to produce equal footcandles for equal watts.

In other words, considering filament and fluorescent direct-indirect luminaires to have identical outputs and distribution would mean that the total watts, and therefore the total heat, could not be the same for a given level of illumination since the fluorescent lamp is generating its lumens (including losses in the ballast) over 100 per cent more efficiently than the filament lamp. It would be astounding if there is not a sufficient difference here to affect air conditioning for a given footcandle level.

It will interest Mr. Tuck to know that much earlier scientific studies comparing seeing ability under natural daylight and the light from tungsten-filament lamps, which showed no appreciable difference, have been checked using the light from Type F lamps and filament lamps. There is thus very little danger that as far as footcandles for filament lamps are concerned fluorescent lamps will ever provide other than cooler footcandles for air conditioning.

DIRECT EVAPORATIVE COOLING FOR HOMES IN THE SOUTHWEST

By A. J. RUMMEL,* SAN ANTONIO, TEXAS

I N certain sections of the Southwest it has been observed that during the past several years greater public interest in the installation of direct evaporative cooling equipment in residences has occurred. In fact, in the areas where this type of comfort cooling is effective, the number of installations has outnumbered all other types of installations by a 10 to 1 ratio.

The mushroom growth of these installations and the increasing public desire for definite information as to just where and how effective and successful these installations are in those sections where the results might be considered doubtful, or on the borderline, resulted in the study reported here.

The history and theory of the natural phenomenon of evaporative cooling is well known, and the study made and reported, was not to determine the merits of the different type units available, or the effect of the shape of pads or blades, dimensions of units, method of water distribution, etc., but only an investigation of the best results that might be expected when the system was installed and operated under conditions considered to be the most effective.

Since there are no generally accepted or definitely established standards for the cooling of residences by simple evaporation, the first step in such an investigation and study was to analyze the climatic conditions existing in different parts of the country, and to analyze the results of research as to temperature, humidity, air motion, etc., as related to human comfort.

An analysis of the climatic conditions of the United States for the past 10 years indicates that there is a strip of approximately 400,000 square miles, about one-eighth of the total area of the United States, extending across parts of Texas, Oklahoma, New Mexico, southern Arizona, Nevada and California, in which the temperature and humidity conditions are such as to make the use of direct evaporative cooling practical. Although in most sections of this strip the use of direct evaporative cooling is effective in varying degrees, there are also sections where the relative humidity would permit the use of these systems but the maximum temperatures do not justify their use.

* Air Conditioning Engineer, San Antonio Public Service Co. MEMBER of ASHVE.

As previously stated, the effectiveness of these installations varies according to the location. For instance, in the territory of Phoenix, Ariz., the record of summer daytime wet- and dry-bulb temperature readings indicates that these units might be expected to be effective 85 per cent of the time during the summer months. In Yuma and Tucson, Ariz., they might be expected to be effective 70 per cent to 75 per cent of the time; in El Paso, Texas, 65 per cent; Oklahoma City, 60 per cent; San Antonio, Texas, 50 per cent, to practically zero per cent as the coast line is approached. The foregoing is on the basis that evaporative cooling is not effective in territories where the wet-bulb temperature exceeds 75 F for more than 5 per cent of the time during the summer.

Recent investigations [1] by the American Society of Heating and Venti-lating Engineers indicate that the optimum effective temperature for com-pletely air conditioned buildings in the Southwest, in which the period of time spent by the occupants is two or three hours, is approximately 73 F. Similar studies [2] conducted in the northern and eastern parts of the United States indicate that the optimum condition in those sections of the country was approximately 71 deg ET. These investigations tend to indicate that indi-viduals in the area where evaporative cooling might be considered effective would be comfortable in conditions approximately 2 deg higher in effective temperature than is commonly thought of as the optimum comfort condition based on the present day published comfort charts.

The question of maintaining comfortable conditions with high relative humid-ities, as is necessary with direct evaporative cooling, almost always invites some criticism. However, recent studies in this connection seem to indicate very little difference in comfort with humidities at the upper limits of the comfort chart. The 1940 edition of the Heating, Ventilating and Air Conditioning Guide has the following to say regarding the limits for comfortable living conditions:

Preliminary experiments at the ASHVE Research Laboratory would seem to indicate no appreciable impairment of comfort with relative humidities as high as 80 per cent provided the effective temperature is between 70 and 75 deg.

......74.5 deg ET and lower, results in satisfactory comfort conditions in the living quarters of a residence and while this condition is not representative of optimum comfort it provides for sufficient relief in hot weather to be acceptable to the majority of users.

Since these conclusions are based on tests made with individuals in the territory where the desired optimum effective temperature is 2 deg lower than in the Southwest it appears logical to suppose that in the territory where direct evaporative cooling is effective the maintenance of indoor effective temperatures in the neighborhood of 76½ deg and lower might be acceptable to the majority of residential users as long as the relative humidity does not exceed the upper limit of the comfort zone. By referring to the probable comfort chart, Fig. 1, as prepared for the Southwest, it will be noted that at

[1] ASHVE Research Report No. 1127—Reactions of Office Workers to Air conditioning in South Texas, by A. J. Rummel, F. E. Giesecke, W. H. Badgett and A. T. Moses. (ASHVE Transactions, Vol. 45, 1939, p. 459.)
[2] ASHVE Research Report No. 1136—Summer Cooling Requirements in Washington, D. C., and other Metropolitan Districts, by F. C. Houghten, Carl Gutberlet and Albert A. Rosenberg. (ASHVE Transactions, Vol. 45, 1939, p. 577.)

76½ deg ET we might expect about 60 per cent of the people to feel comfortable in an air movement of from 15 to 25 fpm.
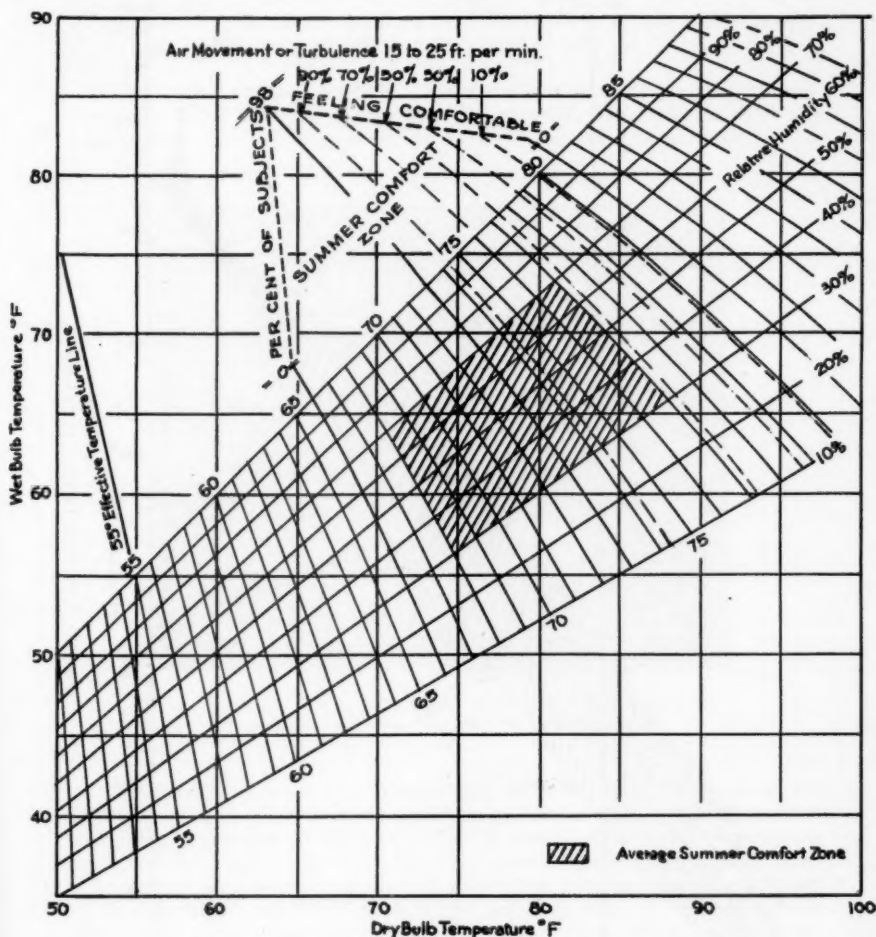
Fig. 1. Probable Comfort Chart for the Southwest

Consider a location in the Southwest where an evaporative cooling installation is operating under conditions of 75 F wet-bulb—the upper limit of wet-bulb where it is considered practicable to use this type of system. If the upper comfort zone humidity limit of 70 per cent is to be maintained with a 75 F

Fig. 2. Effective Temperature Chart

wet-bulb, the dry-bulb temperature will be 83 F.   By referring to the accompanying comfort chart, Fig. 1, it will be noted that under conditions of an air movement of 15 to 25 fpm—the average air movement in most air conditioned installations—and wet- and dry-bulb temperatures of 75 and 83 F, respectively, the effective temperature would be 79 deg.   At this condition only 25 per cent of the occupants might be expected to feel comfortable.

Since it is not possible to obtain a lower dry-bulb temperature and still stay within the upper limit of relative humidity, the only alternative is to obtain a reduction in effective temperature by means of increased air movement.

FIG. 3. DUCT WORK IN ATTIC AND OPENING IN CENTRAL
HALLWAY THROUGH WHICH AIR ENTERS ATTIC

It has been determined that in completely air conditioned buildings, if the air temperature is below 80 F dry-bulb, air movements exceeding 25 fpm have a tendency to produce a feeling of draft.   However, when the air temperature is above 80 F, higher air velocities are possible without a feeling of draft and are desirable.   By referring to Fig. 2 it will be noted that under conditions of 75 F wet-bulb and 83 F dry-bulb, to obtain an effective temperature of 76½ deg, it would be necessary to increase the air movement to approximately 200 fpm.

In an effort to determine whether or not home owners would be comfortable and satisfied under conditions of such high air movements, numerous investigations and personal interviews were made.   In general, it was found that with the most common type of installation, where the air discharge was installed so that the air was introduced into the living quarters in a horizontal direction

through a grille, the main objection was uneven air distribution. With this arrangement, when standing in the path of the air discharge a very high velocity of air was encountered while when out of the path of the air discharge the air movement was insufficient for comfort.

After a careful study of various reactions and objections had been made, an installation was planned in an effort to overcome as many of these objections as possible, and one that would result in the maximum comfort possible

FIG. 4. CEILING PLAQUE THROUGH WHICH AIR WAS INTRODUCED INTO LIVING ROOM

when operating under conditions close to the upper wet-bulb limit the greater part of the time.

The system installed for test purposes was made in a five-room frame cottage located in San Antonio, Texas. The installation was made so that the system could be operated with the windows and doors open or closed, with individual air supplies to each room, and with a humidistat to limit the moisture that would be added to the air.

The evaporative cooling unit itself was installed in the attic of the home as shown in Fig. 3. Individual galvanized iron ducts were run to the center of each room, terminating at the ceiling plaque, as shown in Fig. 4. In the ceiling of the central hallway there was installed a 3 ft by 6 ft exhaust grille having a net free area of 12.46 sq ft and in the kitchen, directly above the gas range, a 10 in. round vent with an area of 0.54 sq ft. Under the eaves of the house the cornice facing was removed and $\frac{1}{4}$-in. hardware screening

installed; also, louvers were installed in the gables at each end of the house. A humidistat which was connected to a water valve supplying the water to the evaporative pads was placed in the central hallway.

The operation of the system was such that, with the windows and doors of the house closed, outside air was drawn through the wetted pads by a blower, after which the air was delivered to each room through the central ceiling plaque shown in Fig. 4.

The installation was made on the basis of a 700 fpm velocity through the ducts, and a 350 fpm discharge velocity from the ceiling plaque. The blower and cooler assembly was mounted on cork in the attic, and the 1 hp motor

FIG. 5.   EXTERIOR VIEW OF TEST HOUSE SHOWING INLET LOUVER
TO COOLER AND EXCESS WATER DRAIN

operating the twin blower was mounted on a floating base. The insides of all ducts delivering air to the rooms were lined with acoustical felt for a distance of two feet from the outlet of the fan. Although all equipment was installed in the attic, there was no objectionable noise or vibration when the system was in operation.

After the air which is drawn into the unit from outdoors has its dry-bulb temperature reduced, it is discharged into the room. When operating with all windows and doors closed, the pressure inside the house is naturally higher than that outdoors. Under these conditions the air, after passing through the rooms, is exhausted through the grille in the central hallway into the attic space and out of the attic, through the openings under the eaves of the house, to the outside.

The installation was completed and put into operation on July 17, 1939. Readings were taken with an anemometer at the exhaust grille in the hallway and kitchen with all windows and doors closed. The average velocity of the air at the hall grille was 388 fpm and at the kitchen grille 420 fpm. At these

velocities the air being discharged into the attic space through the hall grille was 6903 cfm and through the kitchen vent 227 cfm, giving a total of 7130 cfm.

Since the volume of the house was 9300 cu ft and that of the attic 3600 cu ft, the air was being changed in the house every 1.3 minutes, and in the attic every one-half minute.

Static pressure readings taken on the inlet and outlet sides of the blower showed 0.12 in. of water and 0.13 in. of water, respectively, a total of 0.25 in. of water.

The flow of water to the evaporative pads was set at 0.78 gpm. The excess water was taken through a drain pipe on the outside of the house, as shown in Fig. 5, to the back of the house, where a garden hose was connected and the water used to water the lawn and garden.

A 1 hp single-phase motor rated at 6.3 amp and 220 volts was used to operate the twin blower. When the system was operating, as stated previously, readings taken showed that the motor was using 948 watts at 6 amp and 232 volts.

Temperature and humidity readings taken inside when the outside dry-bulb temperature was 94 F and the outside wet-bulb temperature 75 F were 83 F dry-bulb and a relative humidity of 70 per cent—the setting of the humidistat. The temperature in the attic during this same time was 87 F.

Readings were taken of the outside air wet- and dry-bulb temperatures as it entered the unit at the inlet louver and also of the air as it was leaving the house through the ceiling grille. These readings were taken under different outdoor weather conditions with the average result that on days when it was cloudy and the dry-bulb temperature was fairly low, the wet-bulb temperature increased approximately 0.6 F after having passed through the house, and approximately 0.8 F on days of sunshine and high outside dry-bulb temperatures but with approximately the same wet-bulb temperature.

On the basis of the increase in wet-bulb temperature of the air after passing through the house, the equivalent heat removal amounts to 1.45 and 1.97 tons of refrigeration, respectively.

In addition, the difference in heat flow from the attic space to the rooms below is greatly reduced. Attic temperatures in the location of this test are often as high as 140 F, and the difference in heat flow when the attic temperature is only 87 F amounts to a reduction of approximately 40,000 Btu per hour or the equivalent of 3⅓ tons of refrigeration.

Air Flow in CFM	Time in Minutes for One Air Change in House	Reaction
7130	1.3	Comfortable
5840	1.59	Fairly comfortable with just a slight sensation of a too humid condition
3380	2.75	Uncomfortable

Similar studies were made when the air flow was reduced until the amount of air leaving the house through the exhaust grilles was 5840 cfm and 3380 cfm.

The setting of the air flow at 7130 cfm and 5840 cfm was permitted to remain for several weeks and expressions of comfort from people who were in the house for long and short periods of time were tabulated.

On opposite page is a summation of the personal reactions to the three air deliveries.

An effort was made to determine the air velocity at different locations in each room. First a direct reading air velocity meter was used, but these readings indicated rather definitely that there was a considerable turbulent action of the air in all parts of the room. At points within one foot of the walls positive and negative readings were observed in three different directions. After it became evident that no definite measure of air movement could be determined with this instrument it was thought that a more average determination could be made by means of the anemometer. After several trial readings were taken, it was determined that the most positive readings could be obtained on the four walls of the room at the breathing level height. Even with these readings it was quite evident that the air movement as recorded was not the true air velocity due to the turbulent action of the air. This turbulent action was quite noticeable, since in standing in any part of the room one could feel air movement on all parts of the body.

From the readings obtained it was found that only by using a Kata thermometer could a true reading of the air movement be obtained and accordingly a set of readings was taken with a Kata thermometer at a level 5 ft above the floor line. The average of these Kata thermometer readings taken at a distance of approximately 24 in. from the wall and the anemometer readings taken directly against the wall are shown in Fig. 6. These readings were taken when the air flow was 7130 cfm.

Another set of readings was taken at a level 3 ft above the floor and these readings showed a velocity about 10 per cent less than at the 5 ft level.

Table 1 on page 391 gives the average air velocity in each room with the corresponding effective temperatures as determined from the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS Comfort Chart.

The results of these readings show that there is a variation in the effective temperature in different parts of the house from a minimum of 76.1 deg, corresponding to an air movement of 220 fpm, to a maximum of 78.5 deg, corresponding to a 42 fpm air movement with an average condition of 77.8 deg, corresponding to a 92 fpm movement.

To obtain a 76½ deg ET with an average air movement of 92 fpm at a 70 per cent relative humidity, the indoor conditions should be approximately 73½ F wet-bulb and 81½ F dry-bulb.

It was noted that for those people who entered and remained inside the house the average condition was comfortable, with an increased comfort noted

Fig. 6. Residential Evaporative Temperatures in Test House with Direct Evaporative Cooling

(continued)

SECTION "B-B"

SECTION "C-C"

SYMBOLS
Δ — KATA THERMOMETER READINGS
● — ANEMOMETER READINGS
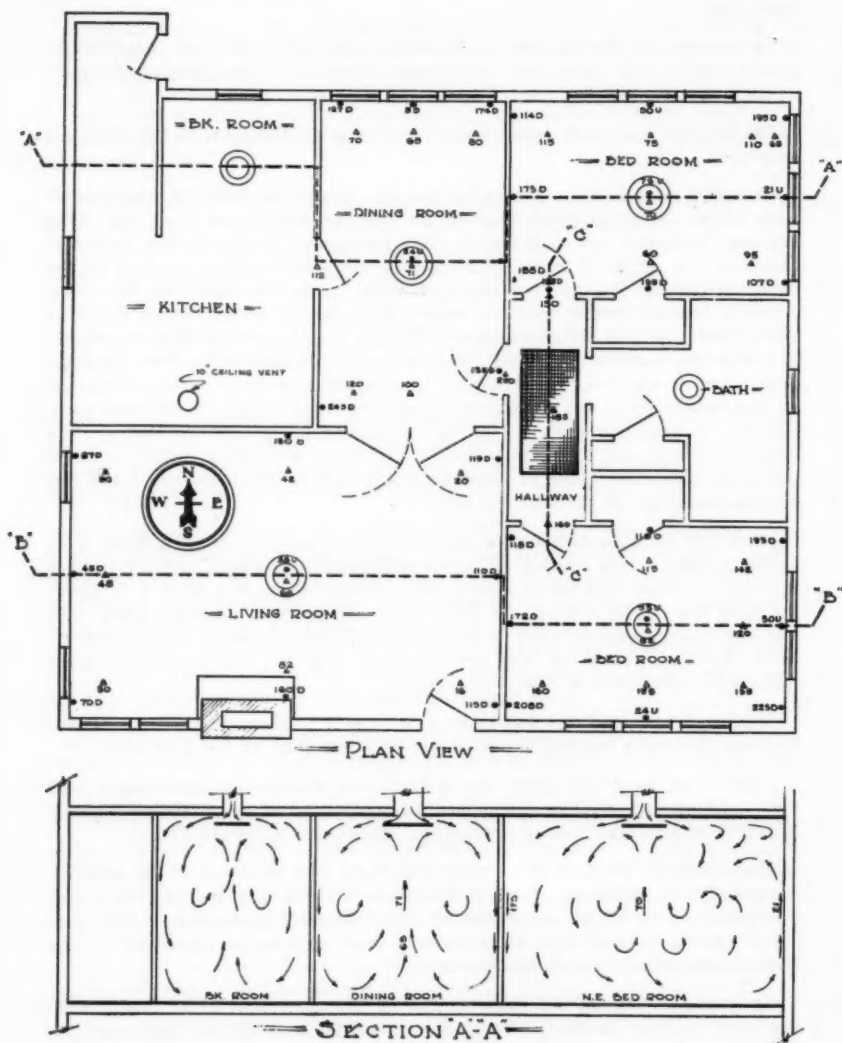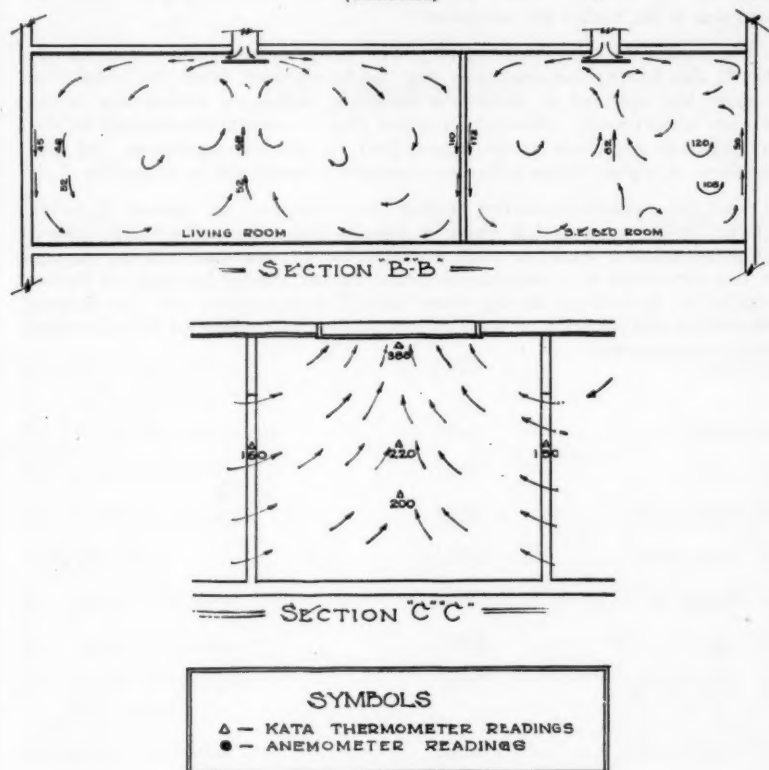
Fig. 6. Residential Evaporative Cooling—Air Velocity
Data

Table 1—Air Movement and Effective Temperatures in Test House with
Direct Evaporative Cooling

Room	Air Movement in Feet per Minute			Effective Temperature 83 deg DB—75 deg WB 70% RH		
	Maximum	Minimum	Average	Maximum	Minimum	Average
Living Room........	90	42	68	77.9	78.3	78
Dining Room........	120	70	92	77.3	78.0	77.8
North Bed Room....	115	60	88	77.5	78.2	77.9
South Bed Room.....	160	82	133	76.9	77.9	77.1
Hall...............	220	160	179	76.1	76.9	76.8

when in the hallway, where the air movement was the highest, with no objection to the higher air movement.

The results of these investigations and tests indicate rather conclusively that in this locality the conditions that can be obtained, when the installation is made and operated as previously described, will prove satisfactory to the average home owner. However, it is felt that the conditions obtained in this investigation are about at the upper limit of comfort conditions, and any condition of higher indoor effective temperature would not be acceptable.

Since this method of comfort cooling has shown such an increase in public interest during the past few years, it appears that more definite and authoritative information should be made available for this phase of comfort cooling. In this connection it is recommended that further studies be made of similar installations in localities having lower wet-bulb temperatures with the ultimate preparation and adoption of some Standards for the Cooling of Residences by Simple Evaporation.

In Memoriam

NAME	JOINED	DIED IN 1940
LESTER P. BRECKENRIDGE (*Honorary Member*)	1920	August 22
JOHN E. EMSWILER	1917	September 23
JAMES J. FINAN (*Life Member*)	1923	July 14
S. FRANKLIN GARDNER	1911	August 29
JOHN MEYERS	1937	June 9
O. C. MONTGOMERY	1933	November 10
EDWARD MOSS	1920	February 5
BENJAMIN NATKIN	1907	January 1
RALPH E. NOLAN	1938	July 28
JAMES W. PARTLAN (*Life Member*)	1916	February 2
JAMES A. PIETSCH	1936	December 15
ROBERT P. RASMUSSEN	1931	February 2
CHARLES RUNKEL	1935	February 27
HAROLD J. SPIELMANN	1933	June 16
BENJAMIN F. STANGLAND (*Charter Member*)	1895	March 26
FRED R. STILL (*Presidential* and *Life Member*)	1904	February 15
WALTER J. WARD	1940	July 6
CLYDE D. WOODBURY	1918	November 15

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